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Life test of axial hydrostatic drive

I. PETRANSKÝ¹, Š. DRABANT¹, A. ŽIKLA¹, P. KLEINEDLER², J. JABLONICKÝ¹,
I. GRMAN¹

¹Slovak University of Agriculture, Nitra, Slovak Republic

²APIS Inc., Turčianske Teplice, Slovak Republic

ABSTRACT: In this paper results of laboratory test of the hydrostatic drive which consists of axial hydrostatic pump type PV 3K-10-033 and hydrostatic motor type MF 3K-10-033 made by APIS Inc., Turčianske Teplice are presented. For the test a biologically degradable plant oil EKOHYD 46 was used to research into a possibility of the replacement of standard mineral oil by plant oil. A special flywheel testing device was designed and built for the test mentioned above. For the tested hydrostatic drive by calculation was determined flywheel of moment of inertia $J_f = 1.3869 \text{ kg/m}^2$. The minimum technical life of the tested hydrostatic drive has been determined 1 million cycles under cyclic pressure loading and the maximum decrease in volume efficiency 20%. The maximum operation pressure in both circuits (A, B) was 42 MPa. The rate of increasing of the operation pressure during the test was 140 MPa/s. The maximum angular velocity clockwise of hydrostatic motor was approximately 44 rad/s and counter clockwise was 38 rad/s. The functional and parametric test and also dimension revise of some functional parts of the hydrostatic pump and hydrostatic motor were accomplished before and also after the test. For the functional and parametric test of the hydrostatic pump and hydrostatic motor a special flywheel testing device was designed and built. Sampling of oil EKOHYD 46 for quality test was carried out at the beginning of the test and every 250, 000 loading cycles. Based on the results achieved during the test it is possible to recommend biologically degradable plant oil EKOHYD 46 for the hydrostatic drive type 3K when the operating conditions according to the manufacturer's prescriptions will be held.

Keywords: flywheel testing device; hydrostatic transmission; hydrostatic drive; mineral oil; organic esters-based oil; EKOHYD 46

At present hydrostatic drive is the most effective transmission of power for mobile machines. Usually internal combustion engine drives hydrostatic transmission which then drives wheels of the machine by gears. Hydrostatic transmission is able to change output parameters of the machine dependently on the load and input parameters proportionally. The disadvantage of the hydrostatic transmission is variable efficiency with respect to the pressure and temperature of the working liquid (RUSŇÁK 1982). Also there is very important to secure cleanness of the working liquid which has a great influence on life of the hydrostatic transmission. Other disadvantage is using of mineral oils in the hydrostatic transmission because in the case of failure escaped oil may cause a pollution of the environment.

With respect to higher demands on the protection of environment new duties for manufacturers and users of mobile machines are determined to provide ecological operation of this machines and devices. For this reason there is necessary to realize an amount of effective activities in all branches of production.

From the environmental, technical and economical point of view there is a possibility to replace mineral oil used in the hydraulic systems of mobile machines by biologically degradable plant oil.

Therefore this problem is also approached by research workers of the Department of Vehicles and Heat Devices, Faculty of Engineering, Slovak Agricultural Uni-

versity, Nitra, in cooperation with APIS Inc., Turčianske Teplice and ZTS TEES Inc., Martin as the Research project VEGA M1/7700/20 which is known under title *Limitation of Negative Influence of Agricultural Mechanisation on the Environment*.

MATERIAL AND METHODS

In this paper a possibility of replacement of mineral oil by biologically degradable plant oil EKOHYD 46 (produced by PETROCHEMIA Inc., Dubová, Slovak Republic) in hydrostatic drive by means of laboratory test is described. Tested hydrostatic drive consists of the axial hydrostatic pump type PV 3K-10-033 and hydrostatic motor type MF 3K-10-033 manufactured by APIS Inc., Turčianske Teplice, Slovak Republic. For the test a special flywheel testing device was used.

Technical data of the tested hydrostatic drive PV 3K-10-033 and MF 3K-10-033 (KOLEKTÍV 1998):

Hydrostatic pump PV 3K-10-033	
- maximum geometrical volume V_{Gmax}	33.3.10 ⁻⁶ /m ³
- displacement of backplate β_G is various in range	<-18°, +18°>
- rated speed n_{Gn}	1,920 rpm
- maximal rated speed n_{Gmax}	3,800 rpm
- minimal rated speed n_{Gmin}	500 rpm
- rated flow Q_{Gn}	63.9 dm ³ /min
- maximal rated flow Q_{Gmax}	126.6 dm ³ /min
- rated pressure p_{Gn}	40 MPa

– permanent operating pressure p_G	42 MPa
– maximum pressure p_{Gmax}	48 MPa
– full – peak pressure p_{Gsp}	52 MPa
– pressure of filling circuit p_p	1.3 + 2.5 MPa
Hydrostatic motor MF 3K-10-033	
– geometrical volume V_M	33.3.10 ⁻⁶ /m ³
– rated speed n_{Mn}	1,920 rpm
– maximal rated speed n_{Mmax}	3,800 rpm
– minimal speed n_{Mmin}	500 rpm
– rated flow Q_{Mn}	63.9 dm ³ /min
– maximal flow Q_{Mmax}	126.6 dm ³ /min
– rated pressure p_{Mn}	40 MPa
– permanent operating pressure p_M	42 MPa
– maximum pressure p_{Mmax}	48 MPa
– full – peak pressure p_{Msp}	52 MPa
– torque at $\Delta p = 42$ MPa and M_{EM}	222.6 N/m

Essential physical and chemical parameters of biodegradable hydraulic liquid EKOHYD 46

– kinematic viscosity at 40°C	45 mm ² /s
– kinematic viscosity at 100°C	9.08 mm ² /s
– viscosity index	210
– point of solidification	-30°C
– flash point	260°C
– acid number	0.9 mg KOH/g
– water capacity	0.1%

This biodegradable hydraulic liquid EKOHYD 46 is made on the base of plant oil and modified by special additives. According to the test CEC-L-33-T-94 the oil mentioned above is biologically highly degradable.

During the test of biodegradable hydraulic liquid EKOHYD 46 following measurements were accomplished:

- functional and parametrical test of the hydrostatic pump and hydrostatic motor before flywheel life test,
- flywheel life test of axial hydrostatic drive,
- functional and parametrical test of the hydrostatic pump and hydromotor after flywheel life test,
- measurement of the physical and chemical parameters of oil EKOHYD 46 during the life test of the hydrostatic drive,
- measurement of wear of some functional parts of the hydrostatic drive before and after flywheel life test.

The designed testing stand for life test of a hydrostatic drive must fulfil the following specification:

- hydrostatic pump speed $n_G = 1,500$ rpm
- pressure variance in main circuit from $p = 0.15$ MPa to $p = 42$ MPa
- rate of pressure increase from 100 MPa/s to 350 MPa/s
- frequency of cyclic loading $f = 0.2$ to 1.25 Hz
- volume of oil tank 100 dm³.

The testing device for life test of the hydrostatic drive is equipped with a control and measurement system. By this systems there is possibility to record time dependent state of pressure in the main and filling circuit, hydrostatic motor speed and control impulse (PETRANSKY et al. 2001). Measurement of the oil temperature in tank is by digital thermometer. The testing device allows measurement in closed circuit when the hydrostatic motor may be tested as a hydrostatic pump. In this case the hydrostatic drive is loaded by pressure valves. A filling of the

oil loss in the main circuit is by gear pump.

The minimum technical life of the tested hydrostatic drive must be 1 million loading cycles and the maximum decrease in flow efficiency 20%. Functional and parametrical test and also the dimensional revise of selected functional parts of the hydrostatic pump and hydromotor were implemented at the beginning and at the end of life test.

Sampling of oil EKOHYD for quality test was carried out at the beginning of the test and every 250 thousand loading cycles.

Dimensional revise of selected functional parts of the hydrostatic pump and hydromotor was implemented by means of coordinate measurement equipment type ZEISS PRISMO 7S VAST. Essential parameters of this measurement equipment are following:

- Measuring range: axle $X = 900$ mm, in axle $Y = 120$ mm, in axle $Z = 650$ mm
- Length accuracy: $U1 = (1.3 + L/350)$ μ m, $U3 = (1.8 + L/350)$ μ m, $U2 = 0.6$ μ m
- Limited temperature range: from 18°C to 22°C.

This measurement equipment is placed in climatization room to secure the required temperature range. For the measurement transducers of the measurement range from ϕ 5 mm to 8 mm are used. This measurement transducers are made of synthetic jewel. Measurement of surface roughness was implemented by equipment type MAHR Perthometer M4Pi.

For measurement of planeness of planar parts a helium lamp with interferential glass was used.

Furthermore following gauges was used:

- Digital length gauge 0 + 150 mm
- Micrometer 0 + 25 mm
- Installometer 8 + 12 mm.

RESULTS AND DISCUSSION

Testing stand was designed on the basis of the specifications mentioned in the previous part of this work. The designed functional chart of testing stand is shown in Fig. 1. The axial piston volume regulated hydrostatic pump HG 1 is connected with a 36 kW power electric motor EM 1 by chain clutch. The pressure energy from the hydrostatic pump is transmitted by main circuit to the axial piston non-regulated hydrostatic motor HM. The main circuit consists of two high pressure hoses TH 1 and TH 2. The torque loading of hydrostatic drive is actuated by a flywheel Z. Filter C 1 is built in the suction pipeline oil and filter C 2 in the cooling circuit of oil. In the tank N a supply of operating liquid is placed, and some amount of heat is also dissipated by tank surface. The required temperature range of working liquid during the test is provided by the cooling circuit, which consists of hydrostatic pump EM 2, filter C 2 and radiator CH.

The operation of hydrostatic drive and flywheel loading device is given by the equation:

$$\frac{V_M \Delta p_M}{2\pi} = (J_M + J_Z) \frac{d\omega_M}{dt} \quad (1)$$

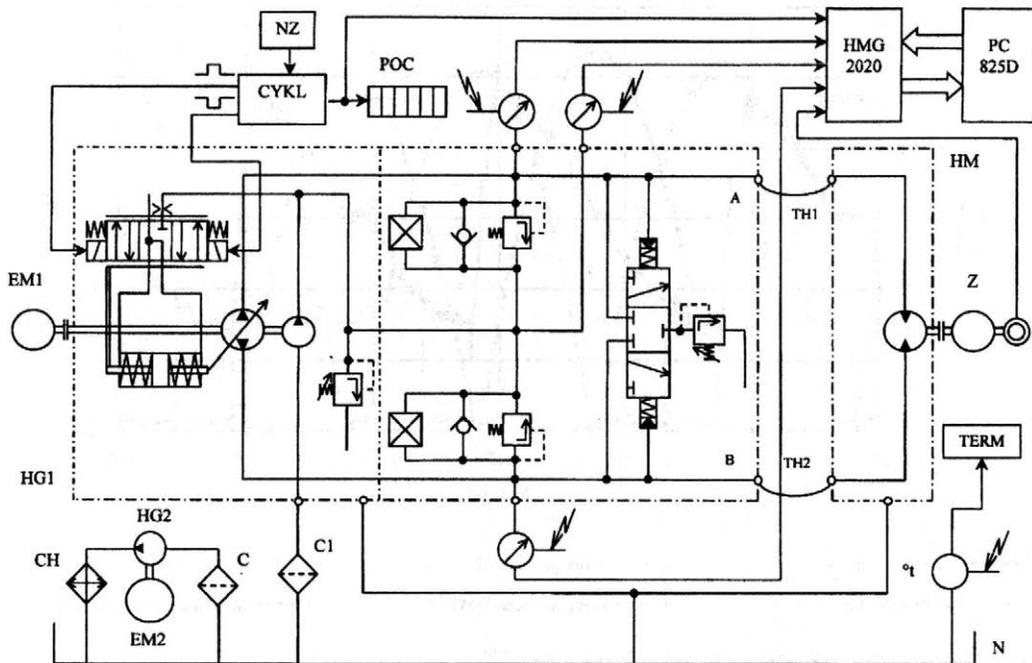


Fig. 1. Functional scheme of flywheel testing device

HG1, HG2 – hydrostatic pump; EM1, EM2 – electric motor; C1, C2 – filter; CH – radiator; HM – hydrostatic motor; TH1, TH2 – high pressure hoses; A, B – main circuit; Z – flywheel; N – tank; ϵ_t – transducer of temperature; TERM – digital thermometer; NZ – power supply unit; CYKL – cyclic pitch controller; POC – counter of cycles; HMG 2020 – measurement and recording device; PC 825D – notebook

where: V_M – the geometrical volume of hydrostatic pump (m^3),
 Δp_M – the loss in pressure in hydrostatic motor (Pa),
 J_M – the hydrostatic motor moment of inertia (kg/m^2),
 J_Z – the flywheel moment of inertia (kg/m^2),
 ω_M – the angular speed of hydrostatic motor (rad/s),
 t – the time (s).

For the constant – displacement hydrostatic motor, the constant of hydrostatic motor can be put into equation (1) as follows:

$$q_M = \frac{V_M}{2\pi} \quad (2)$$

and after mathematical adaptation, the pressure change may be expected as:

$$\Delta p_M = \frac{J_M + J_Z}{q_M} \cdot \frac{d\omega_M}{dt} \quad (3)$$

Equation (3) shows that the pressure change is directly dependent upon the moments of inertia J_M , J_Z and the angular acceleration ϵ_M .

For the variable displacement of the hydrostatic pump MF 3K-10-033, the constant factor q_M is $5.3 \cdot 10^{-6} m^3/rad$ and the moment of inertia J_M is $0.00433052 kg/m^2$. The moment of inertia of flywheel J_Z may be calculated by the following equation:

$$J_Z = \frac{q_M \Delta p_M}{\epsilon_M} - J_M \quad (4)$$

For $q_M = 5.3 \cdot 10^{-6} m^3/rad$, $\Delta p = 42 \cdot 10^6 Pa$, $\epsilon_M = 160 rad/s^2$ and $J_M = 0.00433052 kg/m^2$ the moment of inertia of the flywheel is $J_Z = 1.3869 kg/m^2$.

The stand control system comprises an electrohydrostatic servovalve and an electronic unit CYKL equipped with a cyclic counter POC. The essential parts of the electronic cycling unit are a timer 555, a binary counter 4,017 and a cut-out relay in positive and negative angles. The electronic cycling unit produces two change-timed rectangular electric pulses (in a 0.5 to 4 s range), each other moved about 180° . The pulses control the electrostatic servovalve. Due to this, the back-plate displacement of hydrostatic pump is provided in positive and negative tracks. The total sum of generated pulses is recorded by the POC counter. Electronic cycling unit and electrohydrostatic servovalve are supplied by a power supply unit NZ.

The basis of the measurement system of testing stand is a measuring device HMG 2020 produced by HYDAC Ltd. Using this measurement system it is possible to record 4 analog signals (a maximum input voltage from 4 to 10 volts or a maximum current input of 20 mA) and one frequency signal from 0.3 Hz to 30 kHz. The analog inputs were used for recording of time – dependent states of the pressure in both lines of main circuit, filling pressure and control pulses of electrohydraulic servovalve. The speed of hydraulic motor was recorded by frequency input. A photoelectric speed sensor FS – 1 (developed and

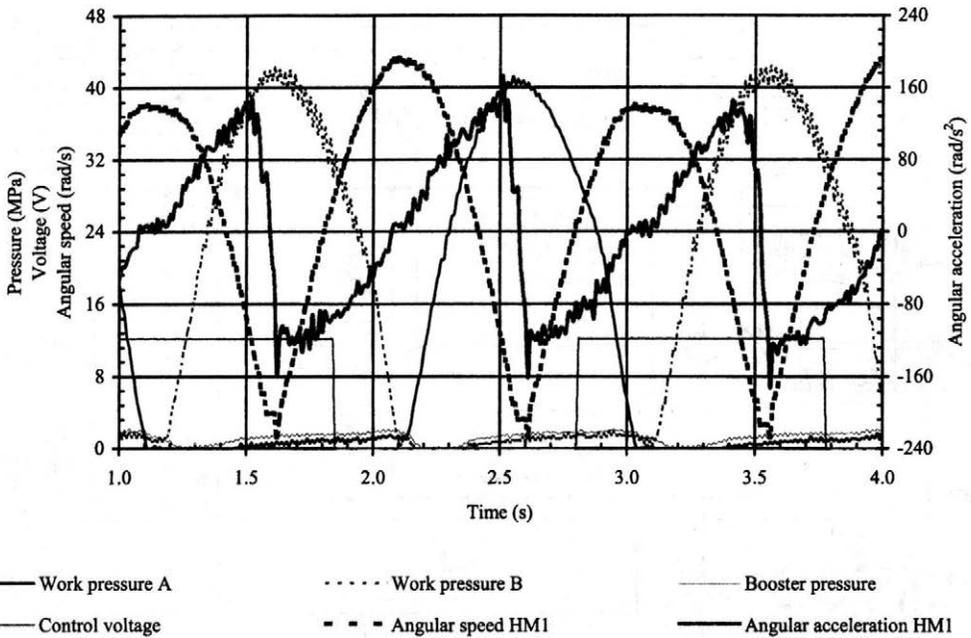


Fig. 2. Time dependent states of measured and calculated values

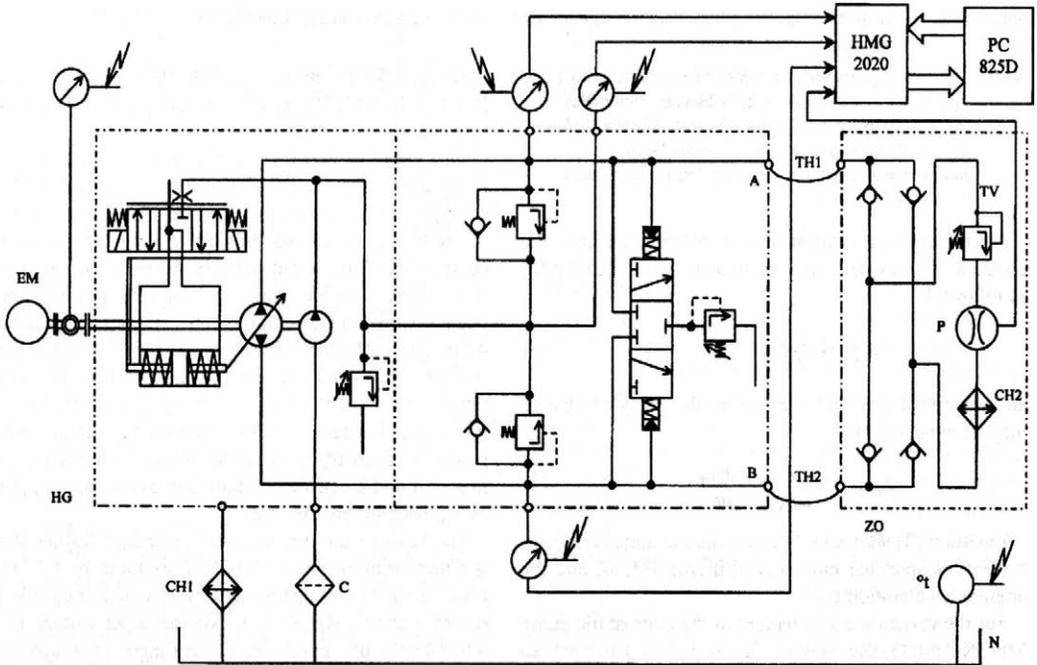


Fig. 3. Function scheme for parametric test of hydrostatic pump

HG – hydrostatic pump; EM – electric motor; C – filter; CH1, CH2 – radiator; TH1, TH2 – high pressure hoses; A, B – high pressure circuit; ZO – loading circuit; N – tank; TV – pressure valve; P – flowmeter; °t – resistance temperature transducer; HMG 2020 – measurement and recording device; PC 825D – notebook

made by the Department of Vehicles and Heating Device, Faculty of Engineering, Slovak Agricultural University, Nitra) was used. The pressure in both lines of main circuit was measured by a pressure sensor HDA 3444-A-600-000 and for determination of liquid temperature, a digital thermometer TEMPOTERM 1 was used.

The measurement equipment HMG 2020 is connected with a notebook Microbook 825D. The measurement system as a whole is controlled by this notebook and a software HMGDESK. After the adjustment of configuration, calibration of sensors, pulse frequency and a number of logging the values of measured units the measurement and recording of the measured values was implemented. It is possible that this makes the time – dependent states

of measured values important. There is a possibility of transferring the measured values into computer for the next treatment by the use of the appropriate software.

The time-dependent states of measured and calculated values during the test of the hydrostatic drive PV 3K-10-033 – MF 3K-10-033 with biodegradable hydraulic liquid EKOHYD 46 are shown in Fig. 2. The operating pressure in main circuit (A, B pipelines) is approximately 42 MPa. The rate of pressure increase is 140 MPa/s. The maximum angular speed of the hydrostatic motor HM 1 clockwise is approximately 44 rad/s and counter clockwise is 38 rad/s. The pressure in filling circuit during the loading cycle is 2.7 MPa. It was found that a time delay between the switching of controlled voltage and the pressure change is 0.27 s. The time-dependent state of angular acceleration of a hydrostatic motor HM 1 was determined by time derivation of the angular speed. The time-dependent state of angular acceleration of the hydrostatic motor has a triangle from and a maximum value is 160 rad/s².

At the beginning and also at the end of the flywheel life test a function and parametric test of the hydrostatic pump and hydrostatic motor was performed. Before the flywheel life test a testing device was designed and built of which function scheme is shown in Fig. 3. The tested hydrostatic pump HG is driven by the asynchronous electric motor EM which is equipped by frequency converter to control speed. Between the electric motor and hydrostatic pump a speedometer is placed. The parameters of the hydrostatic pump were measured in closed circuit which consists of hydrostatic pump HG and loading circuit ZO. A loading device consists of one-way flow rectifier which is equipped with four one-way valves, pressure valve TV, flowmeter P and radiator CH2. By the pressure valve TV a loading pressure in the circuit A or B may be adjusted. The tested hydrostatic pump is connected with the loading device by high-pressure hoses TH1 and TH2. Measurement of speed, temperature and pressure is the same as is shown in Fig. 1. The flow was measured by the flowmeter EVS 3100-3 HYDAC with measuring range from 15 to 300 dm³/min.

Designed and built measuring device for the functional and parametric test of the hydrostatic motor is shown in Fig. 4.

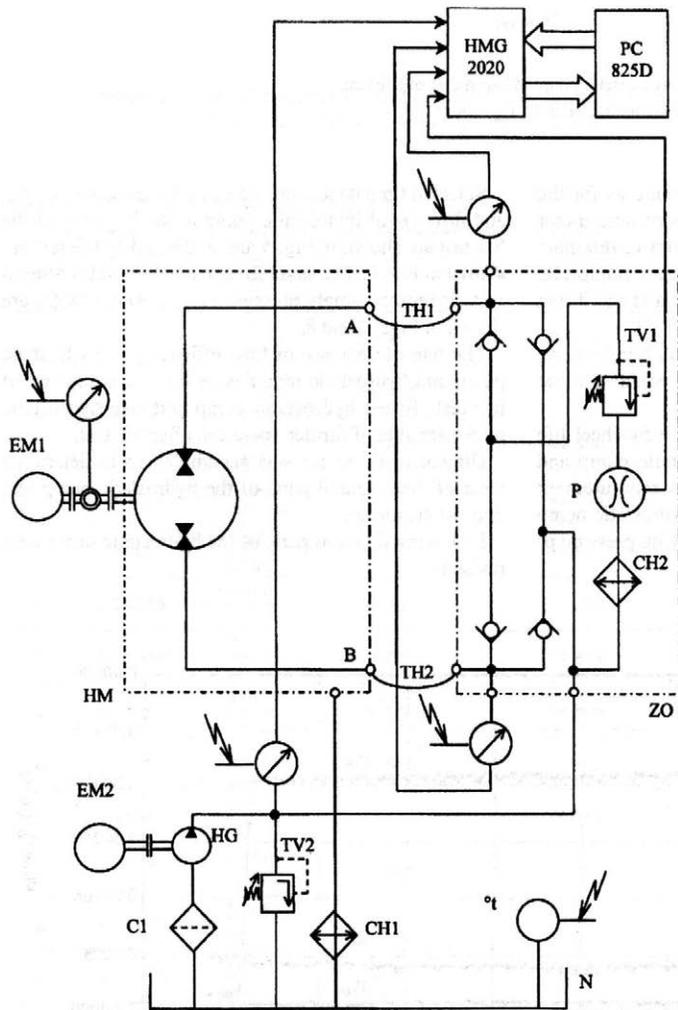


Fig. 4. Scheme of testing device for function test of hydrostatic motor
 HG – hydrostatic pump; EM1, EM2 – electric motor; C1 – filter; CH1, CH2 – radiator;
 TH1, TH2 – high pressure hoses; A, B – high pressure circuit; ZO – loading circuit;
 N – tank; TV1, TV2 – pressure valve; P – flowmeter; °t – resistance temperature transducer;
 HMG 2020 – measurement and recording device; PC 825D – notebook

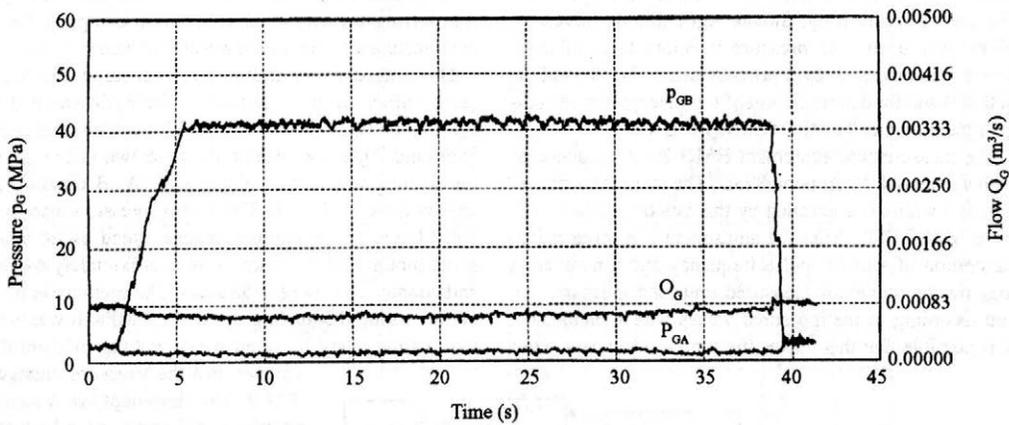


Fig. 5. Time dependent state of parameters of hydrostatic pump at beginning of life test
 p_{GA} – pressure in main circuit A; p_{GB} – pressure in main circuit B; Q_G – flow

In this case the closed circuit is the same as for the test of the hydrostatic pump and the hydrostatic motor makes as a hydrostatic pump. The oil loss in the main circuit is substituted by means of the gear pump HG which is driven by electric motor EM 2 and the filling pressure is controlled by pressure valves TV2.

In this case the measurement of speed, temperature, pressure and flow is the same as for the parametric test according to Fig. 3.

At the beginning and at the end of the flywheel life test essential parameters of the hydrostatic pump and hydrostatic motor were measured. This measurement was accomplished at various speed of hydrostatic pump n_G , hydrostatic motor n_M and also at carrying pressure of high pressure circuit A or B.

Selected time dependent states of the pressure p_{GA} , p_{GB} and flow Q_G of hydrostatic pump at the beginning of the life test are shown in Fig. 5 and at the end of life test are shown in Fig. 6. The same time dependent states of hydrostatic motor, namely pressure p_{MA} , p_{MB} and flow Q_M are shown in Figs. 7 and 8.

The rate of decrease of flow efficiency of hydrostatic pump and hydrostatic motor is below 5% at the end of life test. Tested hydrostatic pump and also hydrostatic motor are able of further operation after life test.

Dimensional revise was accomplished to determine wear of the essential parts of the hydrostatic pump and hydrostatic motor.

Following function parts of the hydrostatic drive were revised:

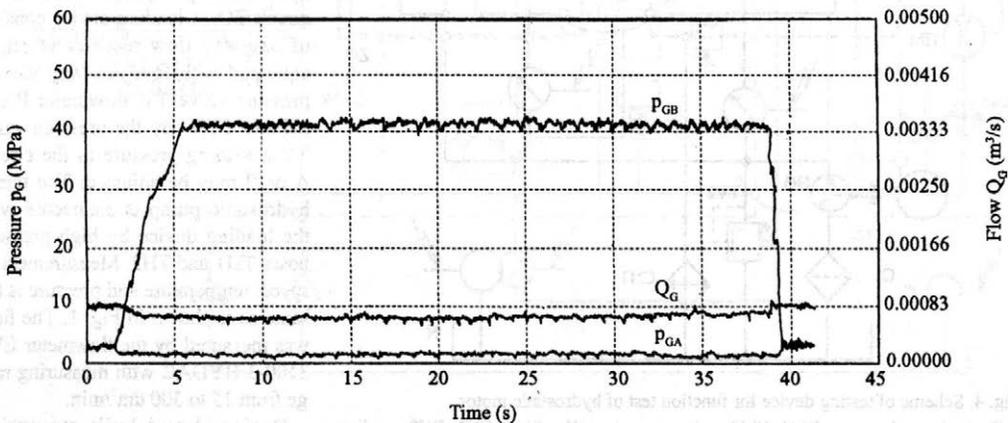


Fig. 6. Time dependent state of parameters of hydrostatic pump at the end of life test
 p_{GA} – pressure in main circuit A; p_{GB} – pressure in main circuit B; Q_G – flow

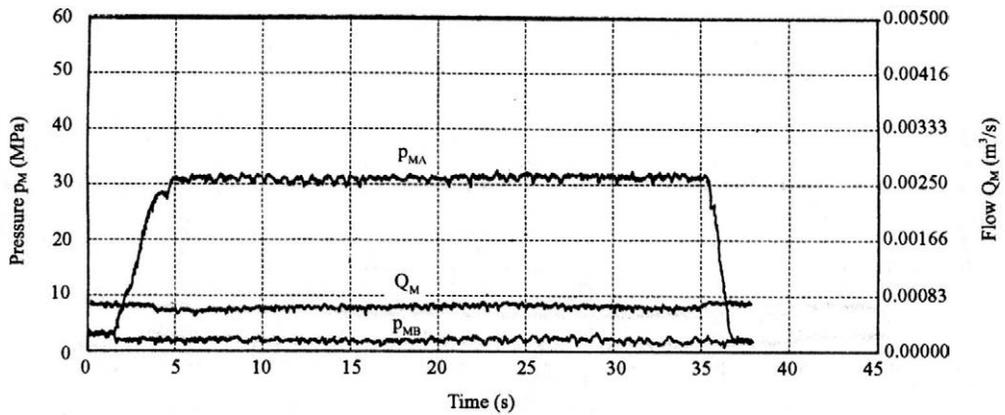


Fig. 7. Time dependent state of parameters of hydrostatic motor at beginning of life test
 P_{MA} – pressure in main circuit A; P_{MB} – pressure in main circuit B; Q_M – flow

Table 1. Measured values of cylinder block

Index	Dimension (mm)	Tolerance (mm)	Measured value (mm)			
			Before life test		After life test	
			Hydrostatic motor	Hydrostatic pump	Hydrostatic motor	Hydrostatic pump
	ϕ 93.7	$\begin{matrix} +0 \\ -0.5 \end{matrix}$	93.473	93.475	93.470	93.470
	ϕ 90.7	$\begin{matrix} +0 \\ -0.5 \end{matrix}$	90.454	90.673	90.460	90.650
	ϕ 44.5	$\begin{matrix} +0.25 \\ -0 \end{matrix}$	44.624	44.556	44.620	44.550
	ϕ 42	$\begin{matrix} +0.16 \\ -0 \end{matrix}$	42.028	42.020	42.030	42.020
	ϕ 40	$\begin{matrix} +0.05 \\ -0.089 \end{matrix}$	39.927	39.923	39.927	39.917
a	ϕ 14.858	$\begin{matrix} +0.011 \\ 0 \end{matrix}$	14.861	14.864	14.862	14.862
b	ϕ 14.858	$\begin{matrix} +0.011 \\ 0 \end{matrix}$	14.860	14.869	14.863	14.862
c	ϕ 14.858	$\begin{matrix} +0.011 \\ 0 \end{matrix}$	14.860	14.866	14.860	14.864
d	ϕ 14.858	$\begin{matrix} +0.011 \\ 0 \end{matrix}$	14.861	14.862	14.865	14.863
e	ϕ 14.858	$\begin{matrix} +0.011 \\ 0 \end{matrix}$	14.862	14.861	14.864	14.862
f	ϕ 14.858	$\begin{matrix} +0.011 \\ 0 \end{matrix}$	14.861	14.865	14.862	14.863
g	ϕ 14.858	$\begin{matrix} +0.011 \\ 0 \end{matrix}$	14.861	14.863	14.865	14.863
h	ϕ 14.858	$\begin{matrix} +0.011 \\ 0 \end{matrix}$	14.861	14.869	14.863	14.862
i	ϕ 14.858	$\begin{matrix} +0.011 \\ 0 \end{matrix}$	14.861	14.861	14.862	14.862
	$^{0.8}\sqrt{\quad}$		0.21	0.37	0.32	0.52
	$^{0.4}\sqrt{\quad}$		0.14	0.12	0.40	0.37
	$^{1.6}\sqrt{\quad}$		1.40	1.36	1.25	1.42

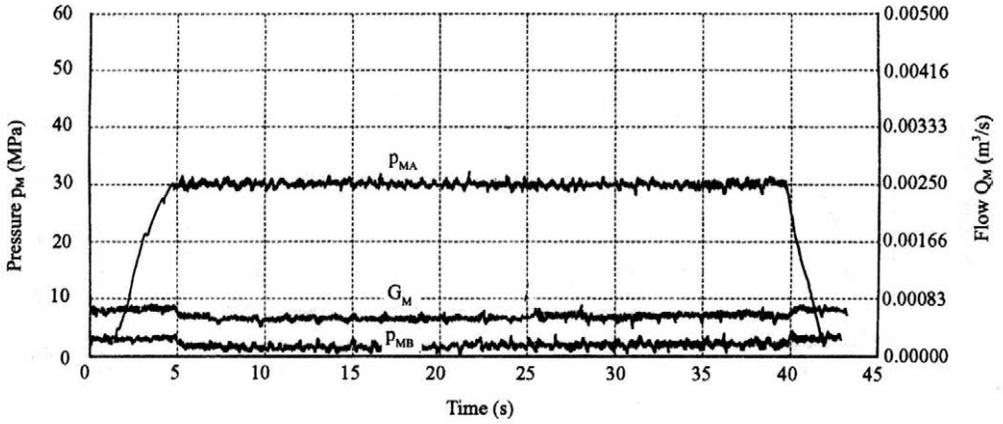


Fig. 8. Time dependent state of parameters of hydrostatic motor at the end of life test
 p_{MA} – pressure in main circuit A; p_{MB} – pressure in main circuit B; Q_M – flow

- block of cylinders,
- sliders,
- dividing desk,
- pressure desk,
- holder of sliders,
- pistons,
- guide of holder and slider,
- sleeve,
- guide of rotor,
- wheel of rotor.

The dimensional revise was accomplished at the beginning and the end of the life test.

The results of dimensional revise of the block of cylinder (Fig. 9) is shown in Table 1. Based on the results it is possible to state that the wear of the function parts mentioned above is pointless.

Sampling of oil EKOHYD 46 for quality test was carried out at the beginning of the test and every 250 thousand loading cycles. The results of the quality test is shown in Table 2.

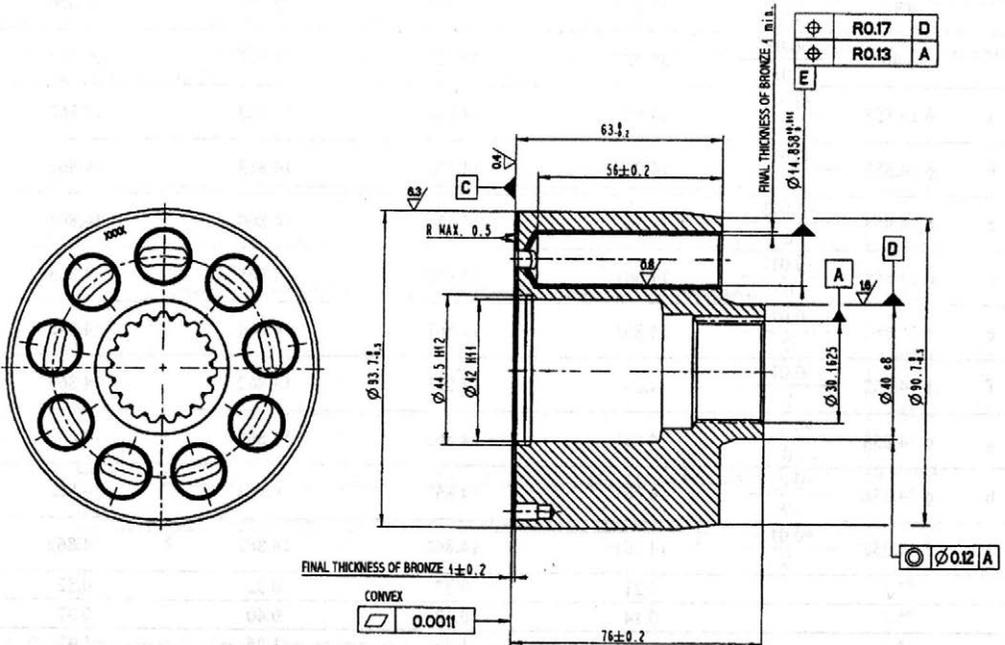


Fig. 9. Dimensions of cylinder block

Table 2. Physical and chemical parameters of biodegradable liquid EKOHYD 46

Physical and chemical parameters	Unit	Measured value				
		0 cycles	2.5.10 ⁵ cycles	5.10 ⁵ cycles	7.5.10 ⁵ cycles	10 ⁶ cycles
Kinematic viscosity at 40°C	mm ² /s	41.63	41.45	41.61	41.98	42.1
Kinematic viscosity at 100°C	mm ² /s	9.18	9.15	9.29	9.17	9.75
Viscosity index	–	211	211	215	209	227
Point of solidification	°C	–32	–34	–34	–30	–30
Flash point	°C	234	224	232	224	220
Acid number	mg KOH/g	0.79	0.83	0.99	0.75	0.84

Based on the results we can state that oil EKOHYD 46 showed a suitable parameters during the life test.

CONCLUSION

In this paper a design of testing device for flywheel life test of hydrostatic drive PV 3K-10-033 and MF 3K-10-033 is presented. The designed testing device which consists of hydrostatic pump type PV 3K-10-033 and hydrostatic motor type MF 3K-10-033 was used to investigate the possibility of replacement of mineral oil by plant oil EKOHYD. The tested hydrostatic drive was made by APIS Inc., Turčianske Teplice, Slovak Republic. The designed testing device was built and long time investigated. It is possible to conclude that the testing device described meets the conditions given by the test standard.

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Životnostné skúšky axiálnych hydrostatických prevodníkov

ABSTRAKT: V príspevku sú uvedené výsledky laboratórneho overenia možnosti náhrady minerálnych olejov rastlinným olejom EKOHYD 46 v hydrostatickom pohone v kombinácii prevodníkov PV 3K-10-033 – MF 3K-10-033, vyrábaných v APIS, a.s., Turčianske Teplice. Overenie vhodnosti použitia biologicky rýchlo odbúrateľnej kvapaliny EKOHYD 46 v hydrostatickom pohone je riešené zotrvačnickovou životnostnou skúškou, pre ktorú bol navrhnutý a zostrojený skúšobný stav. Pre uvedený hydrostatický pohon bol výpočtom určený zotrvačník s momentom zotrvačnosti $J_z = 1,3869 \text{ kg/m}^2$. Technický život pri cyklickom tlakovom namáhaní musí byť minimálne 1 000 000 cyklov, pripúšťa sa zníženie prietokovej účinnosti maximálne o 20 %. Definované podmienky pre návrh skúšobného stavu boli splnené. Pracovné tlaky v oboch vetvách (A, B) dosahujú maximálnu hodnotu 42 MPa. Rýchlosť stúpania tlaku je 140 MPa/s. Maximálne hodnoty uhlovej rýchlosti hydromotora pri pravotočivom smere dosahujú hodnoty približne 44 rad/s a sú väčšie než v smere ľavotočivom, kde dosahujú hodnoty približne 38 rad/s. Funkčné skúšky, parametrické skúšky a rozmerová kontrola niektorých funkčných častí hydrogenerátora a hydromotora boli uskutočnené pred životnostnou skúškou a po tejto skúške. Pre funkčné skúšky a parametrické skúšky hydrogenerátora a hydromotora boli navrhnuté a zostrojené skúšobné stavy. Odoberanie vzoriek oleja EKOHYD 46 pre vyhodnocovanie zmeny fyzikálno-chemických parametrov robilo sa pred meraním a po každých 250 000 cykloch. Na základe výsledkov vykonaných skúšok a meraní je možné odporučiť biologicky rýchlo rozložiteľnú kvapalinu EKOHYD 46 pre hydrostatické prevodníky typu 3K za podmienok dodržania predpisu pre prevádzkovanie.

Kľúčové slová: zotrvačnickový skúšobný stav; hydrostatické prevodníky; hydrostatický pohon; minerálny olej; rastlinný olej; EKOHYD 46

Corresponding author:

Prof. Ing. IVAN PETRANSKÝ, DrSc., Slovenská poľnohospodárska univerzita, Mechanizačná fakulta, Trieda A. Hlinku 2, 949 76 Nitra, Slovenská republika
tel.: + 421 37 772 21 90, fax: + 421 37 741 70 03, e-mail: stefan.drabant@uniag.sk

Energetic requirements of flame weed control

M. MOJŽIŠ

Slovak University of Agriculture, Nitra, Slovak Republic

ABSTRACT: The flame treatment with LPG-burners is completely dominant and today almost the only thermal method used commercially. The method is presently used most in alternative cropping of vegetables and in management of hard surfaces in urban areas. In this paper the results of field experiments with flame treatment of plants under controlled condition are described. The results of flame treatment indicate that weeds effect depending on weeds species, stage of development of weeds and burner parameters. The main parameter for following-up of effectivity of flame treatments in onion was the hectar consumption of gas, which was obtained by changing of speed of flame weeder and changing of gas pressure. The dominant weed species in our trials were Wild oat (*Avena fatua* L.) and Wild radish (*Raphanum raphanistrum* L.).

Keywords: onion; flame weeding

In order to meet demands for reduced chemical control there is a need of rational non-chemical weed control methods. In onion production there is a need for large weed control inputs owing to the weak competitive ability of the onions. Growing onions without herbicides has hitherto required large amounts of hand weeding in the rows, even when very accurate mechanical row-weeding is done. The labour input is frequently the largest cost in production of onions and other root-crops under organic farming condition.

Experiments in other countries have demonstrated that in set onions almost the entire weed control during the cropping period can be done by flaming (VESTER 1986). In seeded onions, on the other hand, large amounts of hand-weeding are required even with flaming. Another investigation compared control strategies in onion (ASCARD 1989) with chemical, thermal and only mechanical weed control and study the weed effect, the labour requirement when hand-weeding and yields. This trial also answered an important question which a combination of thermal and mechanical weed control can replace herbicides. PARISH (1989) conducted laboratory tests on ryegrass and mustard plants and he found that the effect of gas burner depended on its design and angle to the horizontal, and from his tests it was determined that the height of the burner above the ground controlled the effect on ryegrass but not on mustard. No experiments have been made to demonstrate how the consumption of LPG influenced the effectivity of flame weeding for the control of different weed species and yield.

The purpose of this investigation was to compare different modes of action of flame-weeder. The main parameter for following-up of effectivity of flame treatment in onion was the stage of development of weeds and the hectar consumption of gas, which was obtained by changing of speed of flameweeder and changing of

gas pressure. The dominant weed species in our trials was Wild oat and Wild radish.

MATERIAL AND METHODS

The field trial was set out as a randomized block of experiment with twelve treatment (Table 1) and four replications. The hectar consumption of gas was calculated from discovered hour's consumption of gas and times of each treatment.

Each plot measured 8×1.5 m in onion (three double rows of onion – 12 m^2). The sets onion (Advancer F-1) was planted by hand to the three double rows into the 1.5 m wide plots. The distance between the rows was 0.4 m with 7.5 cm distance in the double row. The weed control effectiveness was monitored at four points in each plot by counting weed numbers at start and five days after treatments were done, using a steel frame

Table 1. Parameters of treatment

Treatment	Speed (km/h)	Gas pressure (MPa)	Gas consumption (kg/ha)
T1	2	0.15	36.2
T2	2	0.20	48.5
T3	2	0.25	60.5
T4	3	0.15	24.1
T5	3	0.20	32.3
T6	3	0.25	40.3
T7	4	0.15	16.3
T8	4	0.20	21.8
T9	4	0.25	27.2
T10	5	0.15	14.5
T11	5	0.20	19.4
T12	5	0.25	24.2

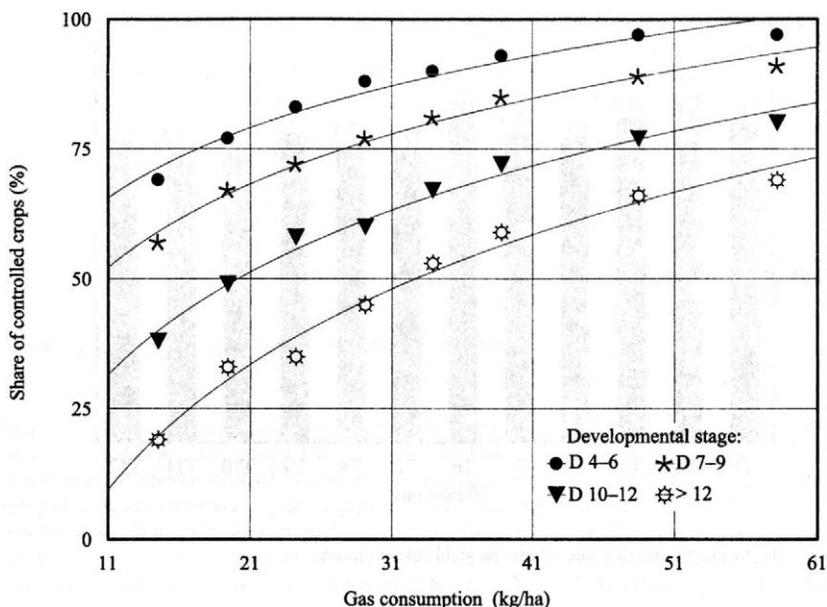


Fig. 1. Influence of changing gas consumption on effectivity of flame weeding on the control of Wild radish

30 × 10 cm, three times in season. The weed species Wild oats and Wild radish was recorded at four different development stages.

The flame-weeding treatments were carried out with a tractor-mounted flame-weeder Reinert – DA211 using a propane gas three times in season, at onion height 4–6, 20–25, respectively 40–50 cm by parameters introduced

in Table 1. The burners were angled at about 45° to the soil and at height 10 cm over the soil. The crop was harvested by hand harvesting a six meters of the inner double row of the onion in each plot.

The results were analysed by the general linear model procedure (GLM) in the SAS system (Statistical Analysis System).

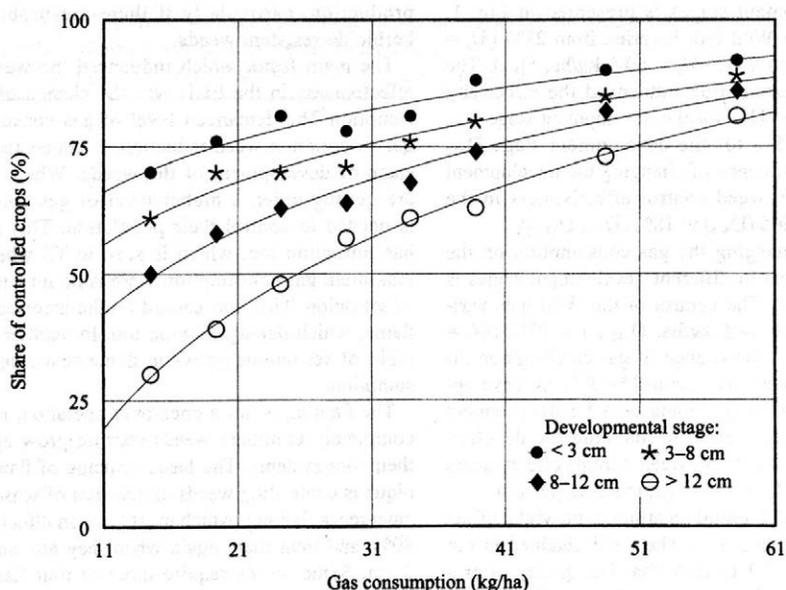


Fig. 2. Influence of changing gas consumption on effectivity of flame weeding on the control of Wild oats

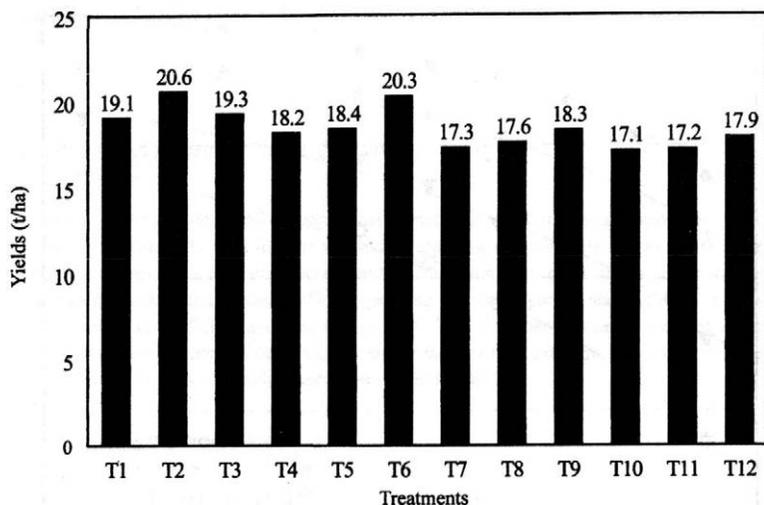


Fig. 3. Influence of changing parameters of flame weeder on yields of set onions

RESULTS

The influence of main factors – the stage of development D and changing a consumption of the gas ΔM_a – was tested on the school farm Bible Hill, Nova Scotia. The four stages of development of Wild radish were recorded in trials D_{4-6} (4–6 true leaves, height 3.5 cm), D_{7-9} (7–9 true leaves, height 7.5 cm), D_{10-12} (10–12 true leaves, height 11.5 cm) and $D_{>12}$ (13 and more true leaves, height 15 cm). The effect of changing the gas consumption on the control of Wild radish in different development stages is presented in Fig. 1. The control of the Wild radish varied from 21% ($M_a = 14.5$ kg/ha, $D_{>12}$) to 93% ($M_a = 60.5$ kg/ha, D_{4-6}). The change of gas consumption influenced the effectivity of weed control by 31% for the development stage D_{4-6} ($P < 0.05$) and 58% for the development stage $D_{>12}$ ($P < 0.05$). The influence of changing the development stage of weeds on weed control effectiveness in the trials varied from 9% (D_{4-6}) to 18% (D_{7-9} , D_{10-12}).

The effect of changing the gas consumption on the control of Wild oats in different development stages is presented in Fig. 2. The control of the Wild oats varied from 31% ($M_a = 14.5$ kg/ha, $D_{>12}$ cm) to 93% ($M_a = 60.5$ kg/ha, $D_{<3}$ cm). The change of gas consumption influenced effectivity of weed control by 33% for development stage $D_{<3}$ cm ($P < 0.05$) and 69% for development stage $D_{>12}$ cm. The influence of changing the development stage of the weeds on weed control effectiveness in the trials varied from 7% ($D_{<3}$ cm) to 21% ($D_{>12}$ cm).

The influence of thermal treatments on yield of set onion is presented in Fig. 3. The yield obtained in our trials varied from 17.1 to 20.6 t/ha. The quality of production obtained in the trials was 1st sort in T1–T6 and T9, 2nd sort in another treatments.

DISCUSSION

Thermal weed control with flaming is, without doubt, a very labour-saving and profitable aid when growing crops without herbicides. Thermal control will be more expensive than chemical control in set onions mainly since the labour cost for hand-weeding will increase. However, the extra cost in organic farming can be paid for by the higher product price. Flaming may also be an interesting alternative or complement to herbicides in conventional forms of production, particularly if there are problems with herbicide-resistant weeds.

The main factor which influenced the weed control effectiveness in the trials was the change of gas consumption. The demanded level of gas consumption to get an optimum weed reduction, is connected with the stage of development of the weeds. When the weeds are getting older, a higher level of gas consumption is needed to control their population. This parameter has limitation too, which is seen in T3 when despite maximum gas consumption there is no maximum yield of set onion. That was caused by the incorrectly set up flame, which damaged onion too. In another treatment yield of set onions grows in dependence on gas consumption.

The flaming is not a once-over operation, nor does it completely eliminate weeds starting grow again from their root systems. The basic principle of flaming technique is controlling weeds on the start of season by pre-emergence flaming, which must have an effect of at least 90%, and treat them again when they are smaller than 5 cm. Some weeds require three or four flamings per year. In this case protection of crops should be used to obtain reasonable yield.

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Energetická náročnosť termickej regulácie burín

ABSTRAKT: Regulácia burín využitím plameňa je dnes jednoznačne najpoužívanjšou termickou metódou z hľadiska praktického využitia. Táto metóda je v súčasnosti aplikovaná najmä pri alternatívnom pestovaní zeleniny, ale tiež pri ošetrovaní mestskej zelene na ťažko prístupných plochách. V práci sú popísané poľné pokusy s termickou reguláciou zaburinenosti pri presne stanovených podmienkach. Dosažené výsledky preukázali, že efektívnosť termickej regulácie burín závisí predovšetkým od ošetrovaného burinného druhu, jeho vývojového štádia a parametrov horáku. Hlavným parametrom pre sledovanie efektívnosti termického ošetrovania v cibuli bola hektárová spotreba plynu, ktorá bola dosiahnutá zmenou pojazdovej rýchlosti a zmenou tlaku plynu. Dominantnými burinnými druhmi v našich pokusoch boli ovos hluchý (*Avena fatua* L.) a reďkev ohnivá (*Raphanum raphanistrum* L.).

Kľúčové slová: cibuľa; regulácia burín plameňom

Corresponding author:

Ing. MIROSLAV MOJŽIŠ, Ph.D., Slovenská poľnohospodárska univerzita, Mechanizačná fakulta, Katedra vozidiel a tepelných zariadení, Trieda A. Hlinku 2, 949 76 Nitra, Slovenská republika
tel.: + 421 37 772 21 88, fax: + 421 37 741 70 03, e-mail: miroslav.mojzis@uniag.sk

Loss of drying medium flow

I. VITÁZEK

Slovak University of Agriculture, Nitra, Slovak Republic

ABSTRACT: In Slovakia (as well as in Moravia) four-channel chamber dryers for drying maize seed were built. In the past the author carried out measurements and analysis of drying medium flow in this kind of dryer. The state with heat exchangers for heat recovery in operation as well as simulation of the state ensuing their disassembly were examined. The method of depth analysis of the flow rate and drying medium loss flows in above mentioned dryer, evaluation of obtained results and proposal of measures for reduction of drying medium loss flows is submitted in this contribution. In measuring the drying medium gauge pressures the resistance of the heat recuperation exchanger installed at the suction side of the fan was proved to be a crucial factor (-1,138 Pa). Recuperators were consequently put out of operation. Following calculations showed high losses of drying medium. Out of the volume flow rate of 7.2 m³/s at the entrance to the drying line L1 there was the volume flow rate of only 5.1 m³/s at the exit, which makes the total loss equal to 2.1 m³/s (29%). The situation is even worse in comparing the drying medium flow rates through chambers in phases F1, F2 and F3. The volume flow rate in phase F2, equal to 4.06 m³/s, which represents only 57.2% of the volume flow rate entering the line, is the lowest of the three. Likewise the volume flow rate through chambers in phase F1 (air from the inlet channel entering direct to the fan) is 5.48 m³/s (77.2%), which show high losses as early as in the inlet channel. Based on these results, the methodology for checking and maintaining air, inlet and outlet flap valves was worked out, and causes of drying medium loss flows were eliminated.

Keywords: seed drying; flow of drying medium; flow losses

Drying is the best way of preservation. It is capable of preserving, and in some cases even improving, the quality of the grain destined for post-harvest processing, and essential for gaining maize seed of high quality. Drying is thus an inevitable part of production of agricultural products such as sugar beet, sunflower and maize seed in particular, harvested in the state of high moisture content.

Providing maize seed of high quality with regard to meeting the demands of post-harvest processing, which this activity requires, led to a need of construction of new dryers. As a result, four-channel chamber dryers with new drying conditions were build (HAVELKA, VITÁZEK 1989).

Post-harvest processing of maize seed differs widely from processing of corn for animal feeding, the main difference being that in the process of drying the whole corn cobs, i.e. the spindle with kernel together are being handled. Moreover, in the process of drying maize seed, great account has to be taken of germination as a basic requirement of preserving high quality of the grain (VITÁZEK 1992). For this reason, the drying medium is only heated up to 30 to 45°C according to the initial dried material moisture content (VITÁZEK 1995a).

Analyses of measured drying medium flow inside a four-channel chamber dryer, its impact on the course of drying process, as well as a proposal of measures for

improving the operation of a dryer are presented in this contribution.

MATERIAL AND METHODS

Tested chamber dryer consists of 16 drying chambers arranged in two lines. The volume of one chamber is 10 t of ear corn. For distribution and discharge of the drying medium, there are four air channels with remote controlled flap valves in each line (Fig. 1). These are: upper inlet channel (UIC), upper (sideways) outlet channel (UOC), lower inlet channel (LIC), lower (sideways) outlet channel (LOC). In each line there is a fan with $Q_A = 11.2 \text{ m}^3/\text{s}$ and $\Delta p_{ev} = 1.8 \text{ kPa}$. An air heater (AH) is used for heating the drying medium which is then forced to the air inlet channel (AIC) by the ventilator. Exhausted drying medium is lead to ambient atmosphere through air outlet channel (AOC). Dryer is equipped with recuperators for recovery of the heat from outgoing drying medium. Drying medium gradually flows through the dried material three times (phases: F1, F2, F3). Fig. 2 shows the layout of the chamber dryer operation system; numbers of chambers in each phase are given below.

A detailed analysis of the flow and parameters of the drying medium inside the drying line L1 was made several times during the season (VITÁZEK 1995b). Values of the drying medium gauge pressure were measured at

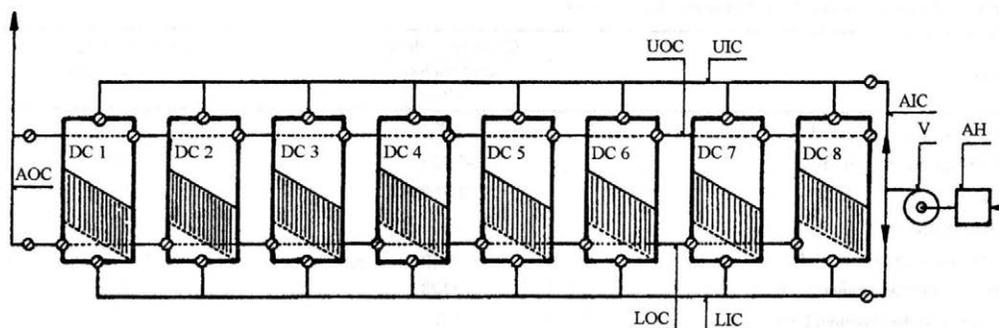


Fig. 1. Construction diagram of one line of the four-channel chamber dryer

selected measuring positions of the line. The positions were: before and after the heat exchanger (HE1), before and after the ventilator (V), in upper inlet channel (UIC), in lower outlet channel (LOC) and in upper outlet channel – exhaust vent (UOC).

State ensuing disassembling of the heat exchangers for heat recovery was simulated by opening the doors in space between the exchangers and fans. Then the measurement of drying medium gauge pressures in selected positions was repeated. Another measured variables were: drying medium volume flow rate at the inlet to the dryer Q_1 and at the exhaust vent Q_2 . During this measurement the dryer was working under following conditions:

- three-phase operation, chambers in operation during each phase:
 - phase F1 – chambers 1, 2;
 - phase F2 – chambers 3, 7;
 - phase F3 – chambers 4, 5, 6;
- conveyance of the drying medium from the fan to the lower inlet channel, i.e. cycle B (direction of drying medium flow changes in all chambers in adjusted time periods, i.e. cycles A and B alter).

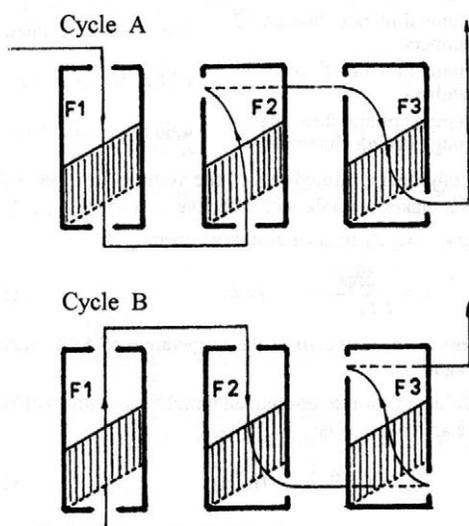


Fig. 2. Diagram of the chamber dryer operation system

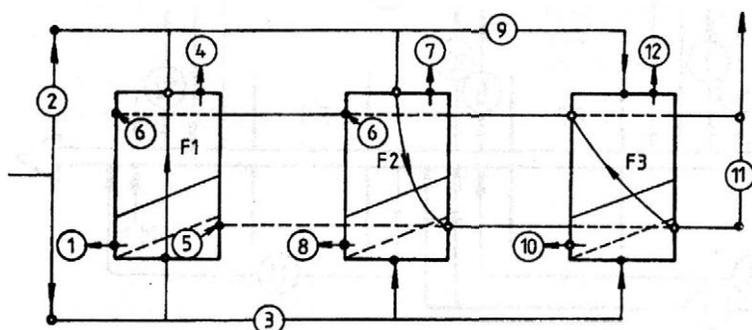


Fig. 3. Diagram of the main technological flow and loss flows of the drying medium in line L1

Table 1. Measured values of the drying medium pressure

Measurement site	Situation with HE1 in operation p (Pa)	Situation with HE1 out of operation p (Pa)
Before heat exchanger HE1	-78.5	0
After heat exchanger HE1	-1,138	0
Before ventilator V1	-1,197	-216
After ventilator V1	+353	+471
Upper air inlet channel UIC	+216	+236
Lower air outlet channel LOC	+128	+118
Upper air outlet channel UOC	+78.5	0

RESULTS AND DISCUSSION

Drying medium gauge pressures were measured at the selected positions in the L1 drying line several times. Values of the pressures measured under B3 conditions are presented as typical ones in Table 1.

First measurement – state with HE1 in operation – is related to standard working conditions. Second measuring – state with HE1 out of operation – represents the state ensuing the disassembling of exchangers for heat recovery.

Fig. 3 shows layout of the L1 line including main technological flow and loss flows of drying medium.

Fig. 4 shows the Sankey's diagram of drying medium flow inside the L1 line. For an easier understanding, there is only one drying chamber drawn in each phase. In real situation, 2 or 3 chambers are in operation. An accurate number of chambers in operation in each phase were used in calculations.

Following values of drying medium volume flow rate inside the L1 drying line were calculated from the values of volume flow rate at the inlet and drying medium pressures obtained by measuring:

Inlet to the L1 drying line	7.2 m ³ /s	measured
Outlet from the L1 drying line	5.1 m ³ /s	measured
Volume flow rate through F1 chambers	5.48 m ³ /s	calculated
Volume flow rate through F2 chambers	4.06 m ³ /s	calculated
Volume flow rate through F3 chambers	4.24 m ³ /s	calculated
Average volume flow rate through drying chambers	4.66 m ³ /s	calculated

Untightness, through which the volume flow rate – Q_z leaks, makes the hole with effective area equal to $\mu_v \cdot S$.

Exit velocity in each discharge opening is:

$$v = \sqrt{\frac{2\Delta p}{\rho_v}} \quad (\text{m/s}) \quad (1)$$

where the ρ_v of the air at the temperature of 40°C equals 1.1 kg/m³.

Volume flow rate through an untightness with an effective area of $\mu_v \cdot S$ is:

$$Q_z = \mu_v \cdot S \cdot v \quad (\text{m}^3/\text{s}) \quad (2)$$

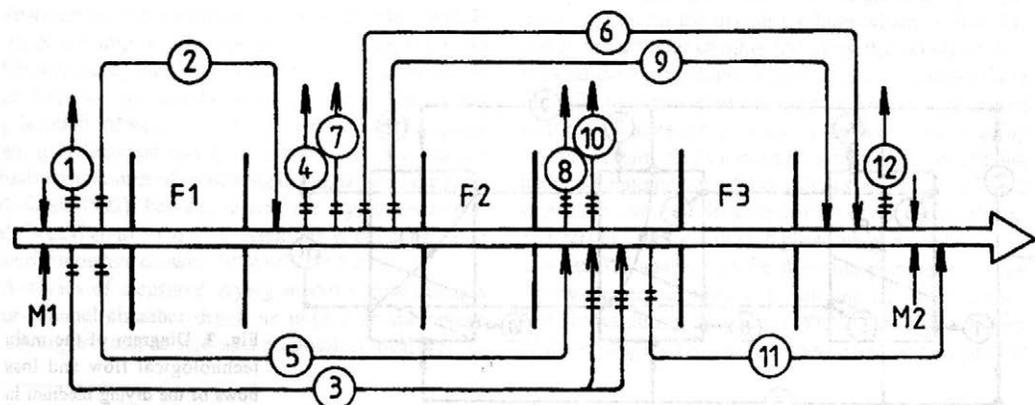


Fig. 4. Sankey's diagram of the drying medium flow in line L1

Table 2. Loss flows in the drying medium

Loss flow No.	1	2	3	4	5	6	7	8	9	10	11	12
Number of lows	2	2	6	2	2	5	3	3	3	3	1	3
Overpressure (Pa)	353	137	226	216	137	137	216	128	137	128	49	78.5
Velocity (m/s)	25.3	15.8	20.3	19.8	15.8	15.8	19.8	15.3	15.8	15.3	9.44	12
Flow rate Q (m ³ /s)	0.37	0.23	0.89	0.29	0.23	0.58	0.44	0.34	0.35	0.34	0.07	0.26

An average value of an effective area $\mu v \cdot S$ of one hole can be calculated from the total loss volume flow rate Q_z measured:

$$Q_z = \sum Q_{zi} = \sum (\mu_{vi} \cdot S) \cdot v_i = \mu_v \cdot S \cdot \sum v_i \quad (\text{m}^3/\text{s}) \quad (3)$$

Solving for $\mu_v \cdot S$, we obtain:

$$\mu_v \cdot S = \frac{Q_z}{\sum v_i} \quad (\text{m}^2) \quad (4)$$

Inserting measured values into Eq. (3) and (4) gives:

$$\sum v_i = \sum n \cdot v = 286.39 \quad (\text{m/s}) \quad (5)$$

$$\mu_v \cdot S = \frac{21}{286.39} = 0.00733 \text{ m}^2 = 73.3 \text{ cm}^2 \quad (6)$$

Table 2 presents drying medium loss flows inside L1 line in the same way as they are presented in Fig. 2 with regard to the number of chambers in operation during each phase. Loss of the drying medium by its escape to atmosphere is represented by loss flows number 1, 4, 7, 8, 10, 11, and 12.

Flow balance in the whole L1 line:

$$\begin{aligned} Q_2 &= Q_1 - \sum Q_{zi} \\ Q_2 &= Q_1 - Q_{z1} - Q_{z4} - Q_{z7} - Q_{z8} - Q_{z10} - Q_{z11} - Q_{z12} \\ Q_2 &= 7.2 - 0.3709 - 0.2903 - 0.4354 - 0.3353 \\ &\quad - 0.3353 - 0.0692 - 0.2628 \\ Q_2 &= 5.10 \text{ m}^3/\text{s} \end{aligned}$$

Flow balance in the drying chambers

Inlet to drying chambers in phase F1

$$\begin{aligned} Q_{F1} &= Q_1 - Q_{z1} - Q_{z2} - Q_{z3} - Q_{z5} \\ Q_{F1} &= 7.2 - 0.3709 - 0.2313 - 0.8906 - 0.2313 = \\ &= 5.48 \text{ m}^3/\text{s} \end{aligned}$$

Inlet to drying chambers in phase F2

$$\begin{aligned} Q_{F2} &= Q_{F1} + Q_{z2} - Q_{z4} - Q_{z6} - Q_{z7} - Q_{z9} \\ Q_{F2} &= 5.48 + 0.2313 - 0.2903 - 0.5783 - 0.4354 \\ &\quad - 0.3470 \\ Q_{F2} &= 4.06 \text{ m}^3/\text{s} \end{aligned}$$

Inlet to drying chambers in phase F3

$$\begin{aligned} Q_{F3} &= Q_{F2} + Q_{z3} + Q_{z5} - Q_{z8} - Q_{z10} - Q_{z11} \\ Q_{F3} &= 4.06 + 0.8906 + 0.2313 - 0.3353 - 0.3353 \\ &\quad - 0.0692 \\ Q_{F3} &= 4.44 \text{ m}^3/\text{s} \end{aligned}$$

Exhaust to atmosphere

$$\begin{aligned} Q_2 &= Q_{F3} + Q_{z6} + Q_{z9} - Q_{z12} \\ Q_2 &= 4.44 + 0.5783 + 0.3470 - 0.2628 = 5.10 \text{ m}^3/\text{s} \end{aligned}$$

Drying medium loss in the dryer

$$Q_z = Q_1 - Q_2 = 7.2 - 5.1 = 2.1 \text{ m}^3/\text{s}$$

With regard to the established facts, it is advisable to take following measures:

1. Keep the door and flap valve gaskets in faultless condition, so as to minimize the drying medium loss due to untightness.
2. Turn the heat exchanger for recuperation of waste heat out of operation. This will result in increase in drying medium volume flow rate by 30% and, consequently, in increase in the capacity of the dryer.
3. Revise the condition of plaster in all air channels and all drying chambers. Repair all discovered faults.

Note: Uselessness of the heat exchangers for heat recovery was also showed by thermal balance, since at the time of measurement the temperature of the drying medium at the exit was equal to the ambient temperature (t_0). So far as the drying of seeds is concerned, the maximum temperature of the drying medium at the entrance is 45°C, and its temperature at the exit is considerably lower, even less than 20°C. Difference in temperatures $t_2 - t_0 = \Delta t$ is very low (5°C, and even lower). In some cases the temperature difference measured Δt was even negative, which shows that the use of recuperators is absolutely inconvenient.

Following the above mentioned measures will enable to reach roughly the same operation parameters as those of the best comparable dryers (HAVEKKA, VITÁZEK 1989).

CONCLUSION

Measuring of drying medium gauge pressures in selected positions of dryer has proved the resistance of the heat recuperation exchanger installed at the suction side of the fan to be a critical place. This was one of the reasons why the recuperators were put out of operation.

The following calculation showed relatively high losses of drying medium inside the dryer. Out of the volume flow rate of 7.2 m³/s at the entrance to the drying line L1 the volume flow rate at the exit was only 5.1 m³/s, which makes the total loss equal to 2.1 m³/s (29.2%). Comparison of the drying medium flow rates through chambers in phases F1, F2 and F3 is also adverse. The volume flow rate in phase F2, equal to 4.06 m³/s, which represents only 57.2% of the volume flow rate entering the line, was the lowest of the three. Volume flow rate through chambers in phase F1 is 5.48 m³/s (77.2), which indicates high losses taking place as early as in the inlet channel. Based on the results, the methodology for checking and maintaining air, inlet and outlet flap valves was worked out, so that the loss flows were minimized and

comparable to those of the best four-channel chamber dryers.

In agriculture the drying, which besides enabling a long-term storage must preserve required characteristic of dried material, is one of the most energy-consuming operations in post-harvest processing. (In drying seeds the quality of final product is determined by its germination.) Improvements in drying equipment and the technology of drying has been reflected in the economical efficiency of the drying process. Finances put to improving the technology and construction, as well as the costs of heating are a part of costs of drying. In the present state of development in technology the three-phase process of drying presents conditions for the best way of drying maize seed. Eliminating the drying medium losses (ca. 30% in our case) will result not only in reducing costs of heating and impacts of combustion flues on the environment near the dryer, but also in consistent keeping of technological rules for drying.

Above mentioned methodology and the analysis and calculation carried out in the work was applied to this

type of dryers in Slovak Republic for the first time. Relatively simple measurements of gauge pressure enabled to carry out this difficult analysis of drying medium flow and its losses.

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Straty pri prúdení sušiaceho prostredia

ABSTRAKT: Na Slovensku (i na Morave) boli vybudované komorové štvorkanálové sušiarne osiva kukurice. V minulom období uskutočnil autor na takejto sušiarňi meranie a analýzu prietoku sušiaceho prostredia. Overovaný je stav so zaradenými výmenníkmi tepla pre spätné získavanie tepla i simulácia stavu, aký by nastal po ich vyradení. Predložená je metóda hĺbkovej analýzy prietoku a strát sušiaceho prostredia na uvedenej sušiarňi, vyhodnotenie zmeraných výsledkov a návrh opatrení na zníženie strát pretiekajúceho sušiaceho prostredia. Ako kritické miesto pri meraní pretlakov sušiaceho prostredia bol zistený odpor rekuperačného výmenníka tepla zaradeného na nasávacej strane ventilátora ($-1\ 138\ \text{Pa}$). Rekuperačné výmenníky boli vyradené z prevádzky. Nasledujúce prepočty preukázali vysoké straty sušiaceho prostredia. Z objemového prietoku na vstupe do sušiacej linky L1 ($7,2\ \text{m}^3/\text{s}$) dosahuje prietok na výstupe $5,1\ \text{m}^3/\text{s}$, teda strata je $2,1\ \text{m}^3/\text{s}$ (29,2 %). Ešte horšia situácia je pri porovnaní prietoku sušiaceho prostredia komorami vo fázach F1, F2 a F3. Najnižší je prietok komorami vo fáze F2 ($4,06\ \text{m}^3/\text{s}$), teda 57,2 % z objemového prietoku na vstupe do linky. Rovnako i prietok komorami vo fáze F1 (vstup vzduchu priamo zo vstupného kanála od ventilátora) je $5,48\ \text{m}^3/\text{s}$ (77,2 %), čo poukazuje na vysoké straty už vo vstupnom kanále. Na základe týchto zistených skutočností bola vypracovaná metodika kontroly a údržby vzduchových, plniacich a vyprázdňovacích klapiek a odstránené príčiny strát sušiaceho prostredia.

Kľúčové slová: sušenie osiva; prúdenie sušiaceho prostredia; straty pri prúdení

Corresponding author:

Doc. Ing. IVAN VITÁZEK, CSc., Slovenská poľnohospodárska univerzita, Mechanizačná fakulta, Trieda A. Hlinku 2, 949 76 Nitra, Slovenská republika

tel.: + 421 37 772 21 88, fax: + 421 37 741 70 03, e-mail: vitazek@mech.uniag.sk

Decreasing energetic demands of vacuum pumps being used in machine milking with utilization of a frequency convertor

E. KUBINA, Š. KOVÁČ

Slovak University of Agriculture, Nitra, Slovak Republic

ABSTRACT: The paper contains energetic and economic evaluation of using a frequency electric voltage convertor being used in regulating engine revolutions to drive vacuum pumps in machine milking. Using a convertor has resulted in a continuous regulation of vacuum pump revolutions and a continuous capacity of vacuum pump in the real time as required by the equipment. By incorporating a frequency convertor engine output decreased on the average from 7,192 W to 3,964 W, by which energy consumption decreased by 45%. SKK 40,680 have been saved for one year when expressed in money. The convertor acquisition price amounted to SKK 65,000. As a result, the investment return is 1.6 year, operating life of the equipment is presumed to be eight to ten years. From the data achieved it follows that using a frequency convertor in the area of driving vacuum pump appears to be highly effective from the energy and economic point of view. By using the convertor compared to a more efficient engine, as much as 70% of savings can be achieved.

Keywords: milking equipment; vacuum pump; frequency convertor; energy saving

In recent times, no attention has been paid to energetic demands in milking equipment because the energy appliances forming a part of their equipment have not been represented significantly by challenging appliances.

The appliances were represented by an electric motor to drive vacuum pumps with power input ranging from 4 to 8 kW, the electric water heaters with power input from 5 to 10 kW, the electric motor to drive milk pump with power input from 0.75 to 1 kW, and the electrical equipment with negligible power input.

At present there are, however, energetic demands, especially in bigger milking parlours which are extended, are playing substantially a greater and more important role. Electric motors to drive highly efficient vacuum pumps have power input of 7.5 to 29 kW. The power input of sanitary solutions heater in sanitary equipment is 7 to 15 kW, the electric motor of milk pump has 1.1 to 2.2 kW and the electrical equipment have remained of few demands despite their big extension and improvement. When summing up these power inputs we will find that if the total power input of the milking parlour was 9 to 24 kW in the past, at present it is from 15 to 46 kW according to the milking parlour size.

Milking equipment is specific by being in operation 365 days in a year and are used 8 to 16 hours daily depending on the number of cows being milked and the number of milking stands available in the milking parlour. Not all energy appliances forming a part of the equipment are in operation the whole time indicated. Electric motors to drive vacuum pumps and the electric motor of the milk pump is running at the time of milking process and sanitation, that is to say, 8 to 16 hours a day. Thus, in a calendar year these appliances are in

operation 2,920 to 5,840 hours. The electric motor of milk pump is working, however, only in certain intervals following the filling up of bulk tank. The sanitary solutions heater is active only in sanitation being carried out before and after milking and lasts 20 to 40 minutes. The heater is heating the required quantity of solution, power input is high because the time needed to heat and maintain temperature affects sanitation time. It is desirable for this time not be too long.

From the above mentioned it follows that the greatest attention deserves the most efficient appliance and the one being in operation for the longest time every day or in a year, such an appliance undoubtedly being electric motor for vacuum pump drive. This appliance has been paid attention to in our paper.

A role of vacuum pump in sanitation and milking is suction, compression and air discharge. In the required spaces of the equipment through this activity is produced vacuum which serves for sucking off milk from the dairy cows' udders and air flow serves for its transportation to the collecting container. The underpressure produced in sanitation is serving to suck the solution and to transport it through all parts of the equipment through which is milk flown into the collecting container. The volume of free air that the vacuum pump is capable of sucking and transporting is indicated as vacuum pump capacity which is indicated as Q_v , and is expressed in units m^3/s or more practically in dm^3/min or l/min , respectively. Part of this capacity under certain conditions of measurement is also called a backup or vacuum pump reserve.

The requirements for vacuum pumps capacity as well as for reserve are determined by ISO Standard 5707

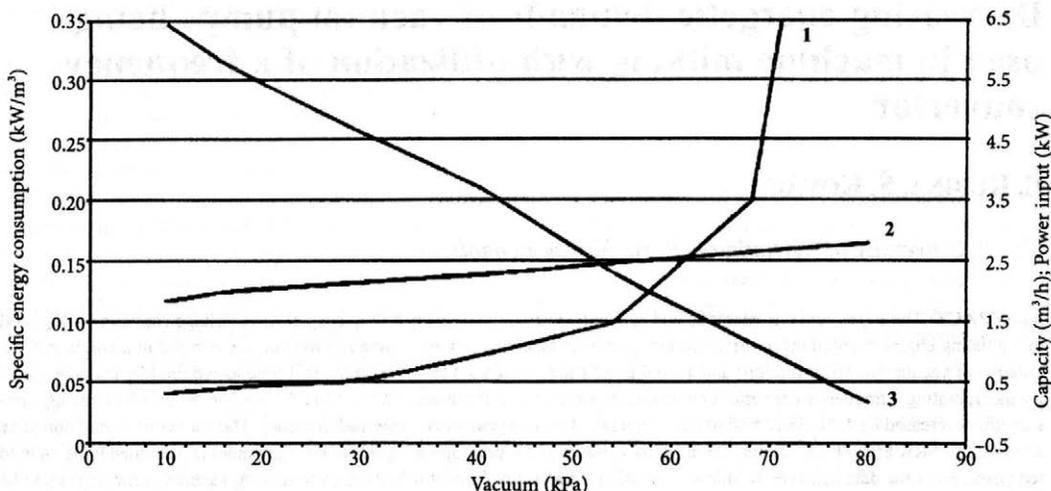


Fig. 1. Relationship of capacity, of power input and specific consumption energy to vacuum of vacuum pump (1 – specific energy consumption, 2 – power input of electric motor, 3 – capacity)

for the whole equipment or for one milking unit. The underpressure value being utilized at milking or sanitation respectively can be adjusted by control valve that is working, regulates the vacuum value by sucking atmospheric air. The requirement for vacuum pump capacity is different at sanitation and at milking itself. The requirement at sanitation is markedly higher when compared with the milking itself. When the vacuum pump meets the requirement for sanitation capacity, then the required capacity at milking is highly exceeded. The required milking capacity can be regulated – modified by rotation frequency of vacuum pump rotor, which is convenient also from the energy point of view. This possibility has been the subject matter of this paper.

Decreasing energetic demands of vacuum pumps can be achieved by three ways:

- decreasing specific consumption of electric energy by proper structure and vacuum pumps installation,
- proper choice of vacuum pump for specific equipment,
- regulating rotation frequency of vacuum pumps following the immediate need of capacity.

Specific consumption has been dealt with by the Research Institute of Farm Machinery Prague-Chodov and the results were presented by KOLÁŘ et al. (1986). Mutual relationship among electric motor driving power, capacity and underpressure are shown in Fig. 1.

Specific consumption belongs among the most important indicators of economy in operation of vacuum pump and is indicated in a table form by producers. Power input required for vacuum pump driving can be found out from torsional moment and rotation frequency as follows:

$$P_p = M_k \cdot \omega = M_k \cdot 2\pi \cdot n, \quad (W) \quad (1)$$

where: M_k – torsional moment (N/m),
 n – frequency of rotating vacuum pump rotor (1/s),
 ω – angular frequency (1/s).

Dependence between efficiency, underpressure and rotation frequency is called a vacuum pump characteristic. Characteristic of a rotary wing vacuum pump is shown in Fig. 2, according to LOBOTKA et al. (1980).

Vacuum pumps with rotary pistons have been largely extended recently because they have several advanta-

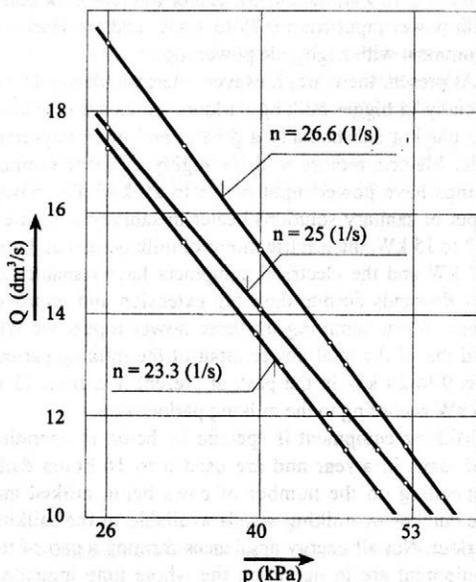


Fig. 2. Vacuum pump characteristic expressing relationship of volume of air sucked to rotating frequency of vacuum pump at different vacuum

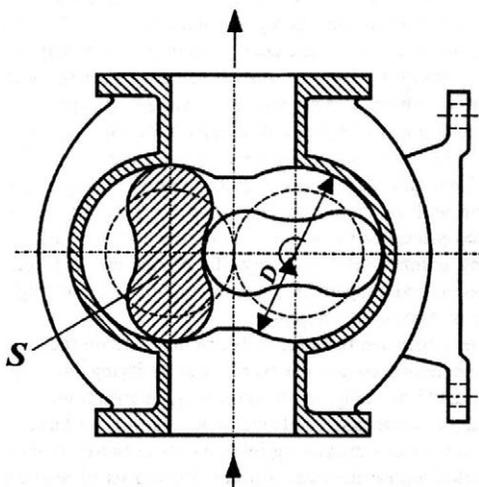


Fig. 3. Vacuum pump with rotating pistons

ges compared with those used up to now. An important advantage is the fact that they do not require lubrication and are running considerably more silently, and so on.

FOGARASI (2000) has been dealing with their construction and use. In order to calculate their efficiency, the following relation is given:

$$V_v = 2 \left(\frac{\pi D^2}{4} - S \right) \cdot b \cdot \frac{n}{60} \quad (\text{m}^3/\text{s}) \quad (2)$$

where: D – the largest piston diameter (m),
 S – front surface of piston (m^2),
 b – rotor width (m),
 n – piston rotation frequency (1/min).

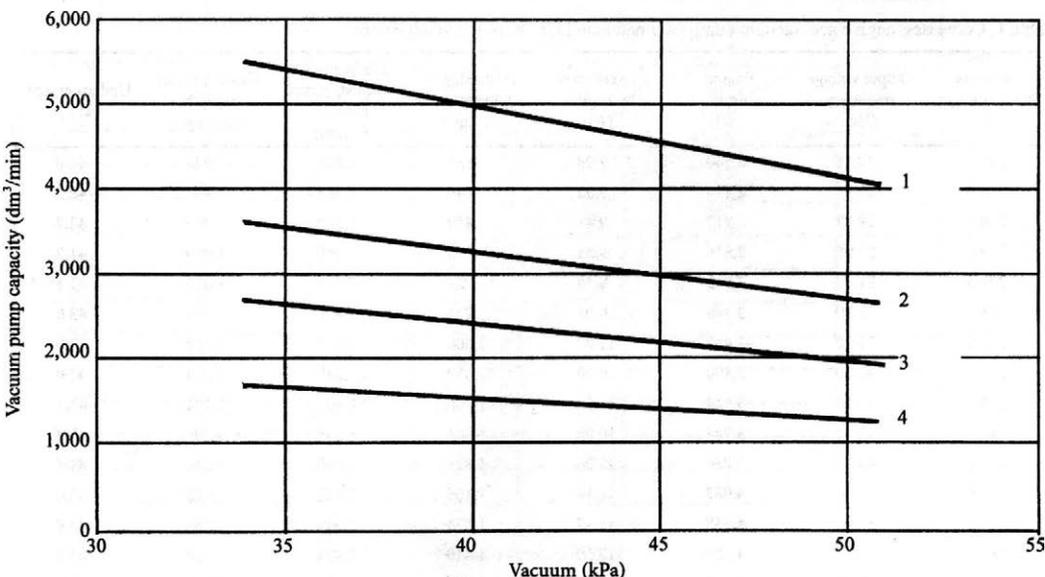


Fig. 4. Power comparison of rotating piston vacuum pump of order BP (1 – BP 400, 2 – BP 200, 3 – BP 140, 4 – BP 100)

To calculate the real – effective efficiency is used coefficient λ conveying efficiency which is ranging from 0.5 to 0.7. The diagram of vacuum pump with rotary pistons is shown in Fig. 3.

The efficiency comparison of rotary piston pumps of BP order being installed by the American company BOU-MATIC into a milking parlour by the same producer are presented in Fig. 4. Vacuum pump of this order, specifically BP-140 is also part of the milking parlour being monitored.

Several authors have been dealing with the development and using frequency convertors, an important one is BULGAKOV (1989) who is stating the possibilities of utilizing electric motors speed control in centrifuges, ventilators, conveyors and the like.

Concrete possibilities of using convertors in vacuum pump driving are presented by ZAJAC (1998). He is stating an example where by using milking parlour with 2×16 boxes for vacuum pump drive with electric motor of 11 kW, as much as SKK 578,160 can be saved up per year, which makes a 60% of saving.

MATERIALS AND METHODS

The experiments and measurements the results of which form the background in the present paper, have been carried out by company BOU-MATIC using a milking parlour with parallel position 2 times 10 milking boxes with a quick leave. The given milking parlour was the first one in Slovak Republic, in which a frequency type convertor ACS 601 has been installed and used. The range of capacity $P = 2,200$ to 3,000 kW, frequency of input voltage was from 43 to 63 Hz and frequency of output voltage of 0 to as much as 300 Hz.

Three phase input voltage was 380–500 V and the range of input voltage was 0 to U input. Nominal input voltage 14/11 A and output voltage 15/11 A. The convertor has three programmable analogue inputs (voltage and two current ones), six programmable digital inputs and three programmable relay outputs.

The setting and survey of parameters measured or regulated can be done by means of a convertor control panel ASC – 601. The numerical display is displaying information concerning condition of three different parameters in actual time which can be changed and others can be calculated from them.

As main parameters are considered: convertor output voltage, frequency of input voltage and engine revolutions. Furthermore, engine power and withdrawn current can be read. Resulting from the principal parameters both pump rotation frequency and efficiency can be determined consequently. The under pressure value was followed by a pointer vacuumeter. In this way it was, at the same time, made possible to read the four parameters, and after changing choice also engine power and current taken by electric motor were allowed to be read as well. The convertor is regulated by a piezoelectric air flow scanner through a regulatory valve with convertor of variable scanned. This facilitates a very quick response (less than 5 m/s).

A part of convertor is also the microprocessor and thermometer digital output of the cooler. For vacuum pump driving was used a three phase asynchronous motor with the basic parameters: producer MEZ Valašské Meziříčí, nominal output 7.5 kW, frequency of supply network 50 Hz, rated speed 1,455 min., rated current $I = 15.1 \text{ A}/8.7 \text{ A}$, power factor 0.82, coverage IP 55. The parameters used with the vacuum pump have been

as follows: model BOU-MATIC AIR-START BP-140, efficiency of vacuum pump at rotor speed 2,920 1/min is 2,800 dm^3/min . It includes a vacuum pump with rotating pistons without lubrication being needed, with gear ratio from electric motor for vacuum pump 1:1.5. 100,000 hours of operation are given for vacuum pump life. At medium operating speed the noisiness is 65 dB. An important part of the vacuum pump is a regulating valve with air flow and underpressure scanner, the so called piezoelectric scanner with convertor of variable being scanned. Before using, the equipment had been adjusted and gauged for specific conditions by an engineer of Agromont company.

The experiments and measurements done on the above mentioned equipment were, after verifying function, conducted by using, in standard way, interruptions over 60 days in the course of one year. We did not have to use any special measuring instruments because the above cited alphanumeric display allowed us to read all data required. As further instruments were used: audio-noise meter, vacuumeter, electronic stop-watch and air flow meter.

Except the standard measurements in the course of sanitation and the milking itself, which totally lasted 5.08 hours, we were reading parameters of indicators investigated. The values measured and read have been processed in a statistical and graphical way.

RESULTS

When adjusting and functional verifying the convertor and scanner we have followed changes of motor and vacuum pump rotations as dependent on the course of air by regulating valve and scanner. In addition, other

Table 1. Convertor, engine and vacuum pump parameters in input testing and adjustment

Convertor input voltage (V)	Output voltage frequency (Hz)	Engine output (W)	Current taken by engine (A)	Frequency of engine rotation (1/min)	Frequency of vacuum pump rotation (1/min)	Vacuum pump capacity (dm^3/min)	Underpressure (kPa)
201.9	29.15	2,540	9.08	839	1,289	971	42.0
210.0	29.11	2,715	9.33	849	1,306	984	42.5
218.0	29.17	2,712	9.41	859	1,320	995	42.7
219.4	29.95	2,836	9.53	898	1,380	1,040	41.9
221.6	31.40	2,942	9.58	924	1,420	1,070	42.1
225.5	33.20	3,146	10.07	975	1,498	1,129	43.0
228.8	33.80	3,487	11.00	1,000	1,537	1,158	43.0
231.1	36.40	3,490	10.90	1,052	1,617	1,218	43.0
232.5	38.40	3,544	11.00	1,100	1,691	1,274	43.1
246.0	42.40	3,743	10.98	1,201	1,846	1,391	43.2
254.8	46.10	3,884	11.00	1,321	2,030	1,530	43.4
264.8	50.00	4,087	11.14	1,406	2,161	1,628	43.4
274.2	54.00	4,358	11.47	1,525	2,344	1,766	43.6
289.2	57.30	4,929	12.30	1,610	2,474	1,864	43.8
306.8	60.20	5,229	12.30	1,700	2,613	1,968	43.9
382.0	64.47	6,616	12.50	1,898	2,917	2,198	44.0

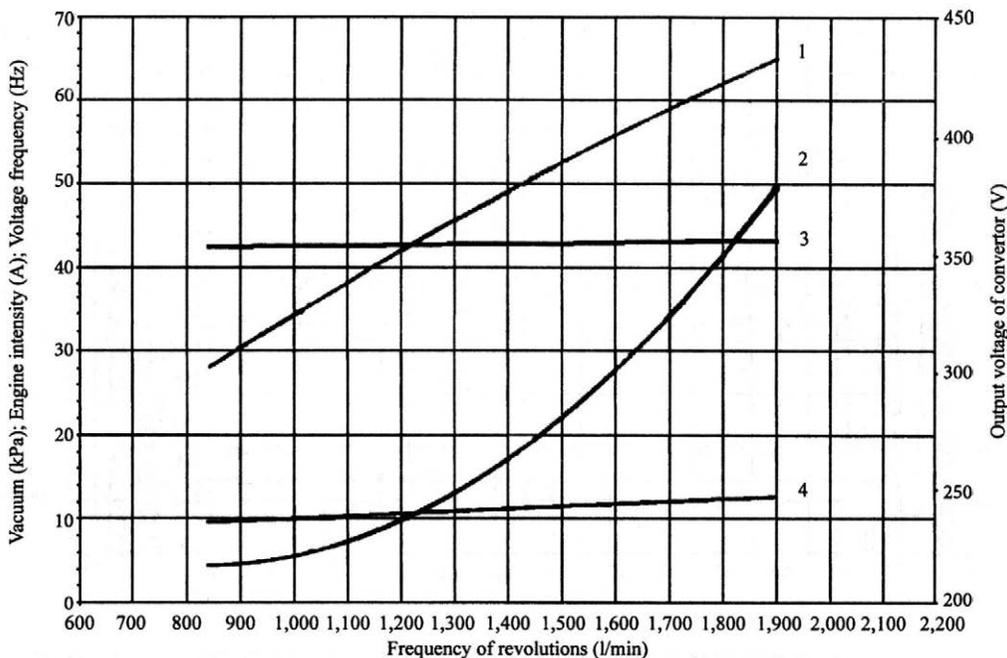


Fig. 5. Graphical illustration of rotation characteristic of converter ACS 600 (1 – voltage frequency, 2 – output voltage, 3 – vacuum, 4 – engine intensity)

parameters which are given in the following Table 1, have been studied. Fig. 5 has been made up as a result of values measured.

From Table 1 and Fig. 5 it may be seen that in the course of sanitation and milking, the electric motor worked at variable revolutions ranging from 839 to

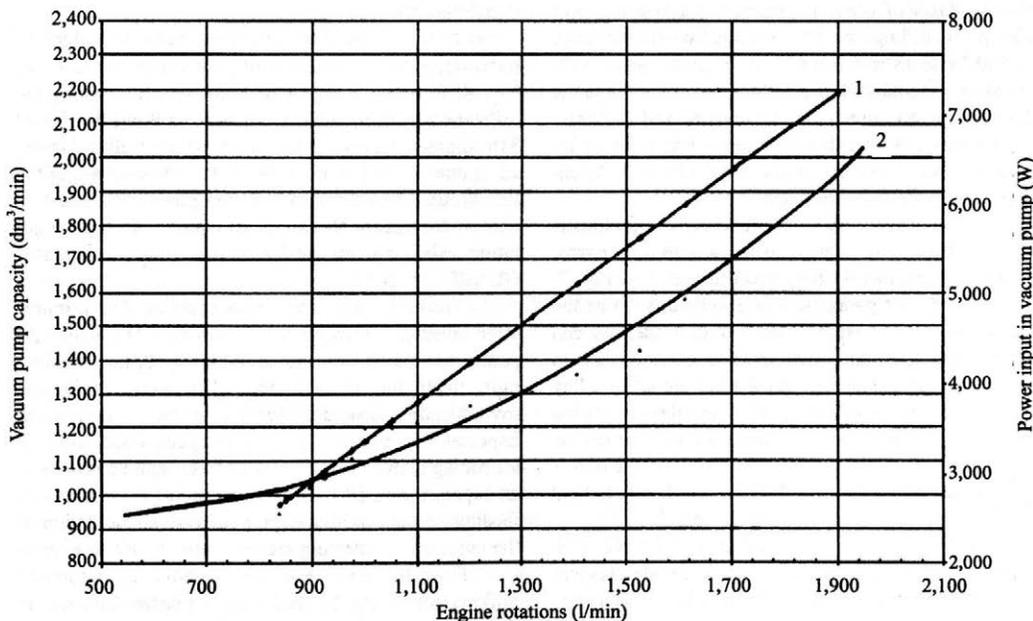


Fig. 6. Vacuum pump capacity and power input of vacuum pump as dependent on engine revolutions (1 – vacuum pump capacity, 2 – power input of vacuum pump)

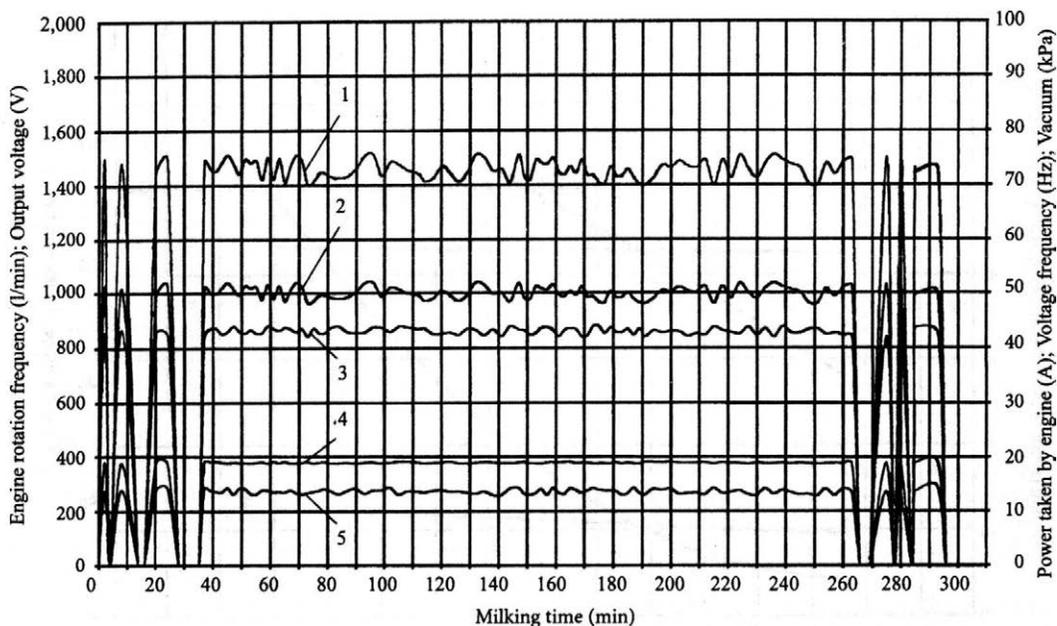


Fig. 7. Diagram of courses of variables measured without regulation by convertor as dependent on sanitation time and proper milking (1 – engine revolutions, 2 – voltage frequency, 3 – vacuum, 4 – output voltage, 5 – intensity taken by engine)

1,898 1/min. Vacuum pump worked with revolutions of 1,289 to 2,917 1/min. These revolutions corresponded with vacuum pump efficiency of 971 to 2,198 dm^3/min . Vacuum as a control parameter was in variation of ± 1 kPa from adjusted nominal vacuum 43 kPa.

On the basis of other parameters studied we have made up Fig. 6. Engine output changed within the range of 2,540 W to as much 6,616 W. Vacuum pump efficiency in a concrete milking parlour corresponded to the efficiency per one unit 48.5 dm^3/min to 110 dm^3/min . The graph in question demonstrates a proportional increase of input power of electric motor drive as dependent on vacuum pump efficiency.

In order to evaluate convertor function and its importance, detailed measures without using a convertor have been made. Graphically they are illustrated in Fig. 7. When the order of parameters is observed, such as the parameter courses as enumerated in Fig. 6, then we can state that engine revolutions are from 0 to 1,455 1/min in the course of sanitation. Zero revolutions are achieved by an engine only when the computer controlling sanitation stops the engine and lets sanitation solution be active. In the course of proper milking, frequency of rotation is slightly varied ranging from 1,400 to 1,455 1/min. In final sanitation the course is similar as in the initial.

Voltage frequency (2) is permanent 50 Hz with a slight variation in network. Observing underpressure value (3) is serving to control its stability. Underpressure was adjusted on a regulating valve to a value of 43 kPa and its fluctuation was within acceptable deviation ± 1 kPa. Output or feeding voltage of electric motor (4)

was 378.9 V on the average. The current being taken by engine on the average was 14.3 A and its deviation was ranging from 0 to 16 A. In the course of the whole utilization, engine power was 7,115.7 W on the average and its deviation was according engine loading ranging from 0 to 7,640.2 W.

Function of convertor and the course of values of parameters measured by using convertor is shown in Fig. 8. Following Fig. 8 one can read that complete utilization of the equipment in one milking has lasted 316 minutes. Of which sanitation before milking lasted 25 minutes with five minutes set to prepare cows of the first group. The activity of vacuum pump and sanitation equipment except the air flow scanner through regulating valve is managed by programmable equipment GUARDIAN II.

The milking itself with breaks lasted 275 minutes after milking sanitation of 11 minutes. The values of parameters as shown and seen in Fig. 8, are considerably fluctuating in sanitation and in very regular and rhythmically repeated cycles also in the milking itself. Especially parameter (2) of engine speed whose curve is ascending above the curve (1) underpressure has marked an expressive oscillation. Also curve of (3) frequency of feeding voltage oscillates proportionally. Oscillation of the curve (4) illustrating current value is not so expressive. From the given Fig. 8 the whole sanitation and milking course can be read with a relative accuracy. In the course of sanitation the engine was started up and stopped according to a programme and its output was utilized from 0 to as much as 56%. Similarly, frequency

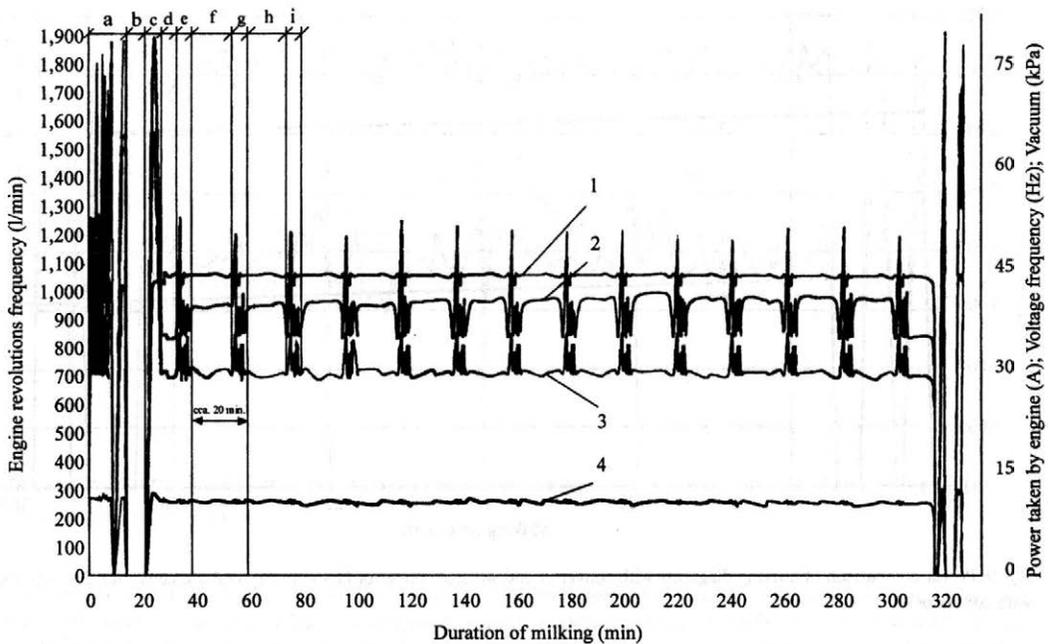


Fig. 8. Graphical illustration of the course of converter parameters measured during sanitation before milking, during milking and sanitation after milking (1 – vacuum, 2 – engine revolutions, 3 – voltage frequency, 4 – intensity taken by engine) a – sanitation before milking, b – break, c – drying of milk conducting lines, d – getting on cows on their milking stands, e – applying teat cups to first group, f – milking, g – removing teat cups, h – changing dairy cows, i – applying teat to second group

of engine revolutions oscillated within the range of 0 to 1,900 l/min.

Proper milking started up by mounting milking sets of the first group in the 31st minute. In consequence of penetrating atmospheric air and higher consumption of underpressure, output voltage increased to 299 V and frequency of output voltage rose to 35–40 Hz which corresponded with engine rotation frequency 1,000–1,100 min. Engine efficiency was utilized to 42–49%. The mounting took 10 minutes.

The mounting having finished, output voltage dropped to 211 V and frequency of output voltage decreased to 29 Hz, which corresponded with the frequency of engine speed 935 min, engine output being utilized to 29%.

When proper milking had finished, milking sets were gradually removed, converter output voltage increased to 373 V, and frequency increased to 44 Hz, with engine rotations increasing to 1,240 min.

The values of parameters are very balanced in all groups of cows.

After all cows had been milked, there was a short six minute break and after sanitation followed which started by sucking off milk residues. Output voltage reached the value of 365 V with frequency being 48 Hz, to which corresponded engine revolutions 1,258 min, with engine output being utilized to 50%.

In Table 2 the courses of engine output during milking with a converter and without a converter are statistically evaluated.

Graphical comparison of efficiency of vacuum pump driving by using no converter and using a converter is presented in Fig. 9. Difference of linear output without using converter and with converter is indicated in Fig. 9 ΔP and reached the value of 2,272 to 3,776 W.

The equipment was utilized daily for 10.1 hours. When no converter was used, electric motor daily consumed 64.8 kWh and with converter 33.0 kWh were consumed. When converter was used daily saving represented 31.8 kWh.

Rate per 1 kWh taken represented SKK 3.50. Consequently, financial cost for electric energy without using converter would amount daily to SKK 226.80 and when converter was used the sum would amount SKK 115. Annual financial costs for energy without converter would amount to SKK 82,782 and when using these would amount to SKK 42,157. Saving expressed finan-

Table 2. Courses of engine outputs during milking

	Output without converter (W)	Output with converter (W)
Mean value	7,192.1	3,963.8
Mean value error	23.0	52.7
Standard deviation	227.5	521.8
Difference max.–min.	881.5	2,386.4
Minimum	6,758.7	2,982.1
Maximum	7,640.2	5,368.0

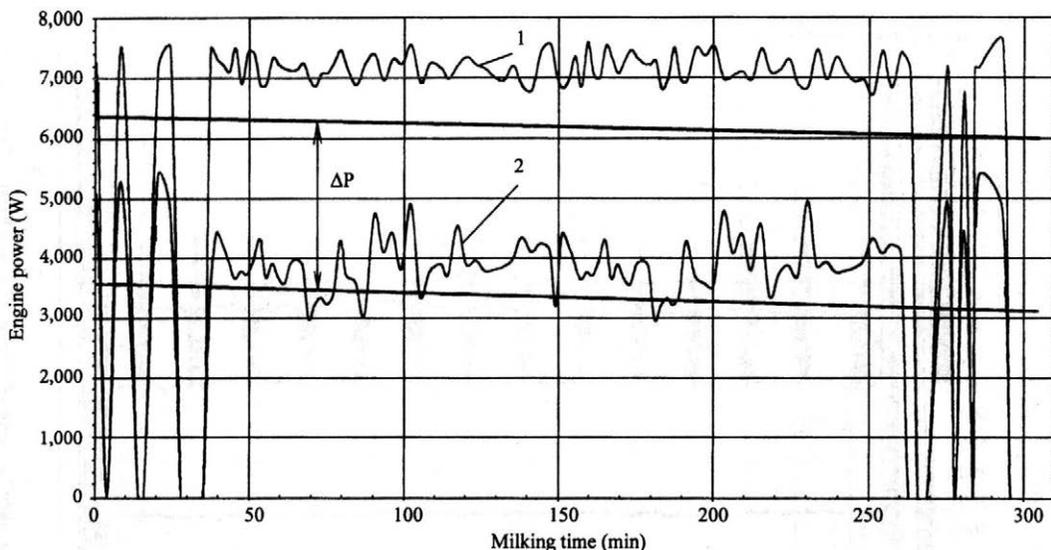


Fig. 9. Power comparison of milking diagrams with converter and without converter (1 – capacity without converter, 2 – capacity with converter)

cially makes then SKK 40,625. In percentage expression the saving for electric energy makes 49%.

Financial costs to equip converter following the seller amount to SKK 65,000. As a result, financial return makes for 1.6 year. The supplier delivering the equipment states their working life to be 12 to 15 years. Even if we consider that their lifetime in our country is generally only eight to ten years, during this time we save electric energy in the amount of SKK 341,000 to 406,625.

To this basic advantage can be added a very low oil consumption to lubricate gears, remarkably improved ecology of using vacuum pump, lower noisiness and totally lower stress of equipment while running in lower revolutions.

DISCUSSION

Energetic demands of vacuum pumps acquires increased importance especially in bigger and higher capacity milking parlours which are intensively extended in the Slovak Republic. A most important energy appliance is represented by electric motor for vacuum pump drive. One of the methods how to effectively manage vacuum pump capacity in actual time is to use frequency and voltage converter. References (ZAJAC 1988 and some others) state potential energy saving of 40 to 60%. This possibility has been confirmed by results we achieved. Higher saving can be achieved by using more efficient vacuum pumps and thereby more efficient driving en-

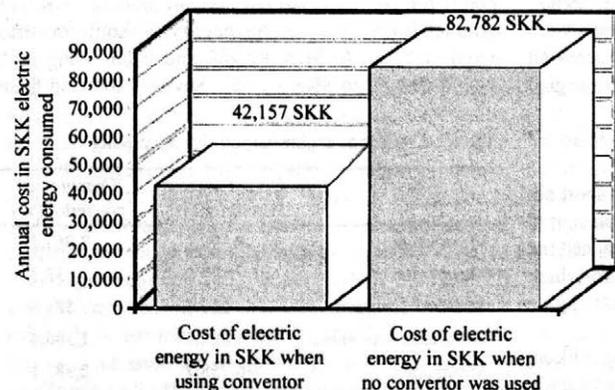


Fig. 10. Financial costs for electric energy consumed per year by using converter and without using converter

gines. The results presented in this paper confirm that it is possible to save energy also in this area of farm production.

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Zníženie energetickej náročnosti vývev používaných pri strojovom dojení s využitím frekvenčného meniča

ABSTRAKT: Príspevok obsahuje energetické a ekonomické vyhodnotenie využívania frekvenčného meniča elektrického napätia používaného na reguláciu otáčok motora na pohon vývevy pri strojovom dojení. Výsledkom využívania meniča je plynulá regulácia otáčok vývevy, a tým i plynulá regulácia výkonnosti vývevy v reálnom čase podľa požiadavky zariadenia. Zaradením frekvenčného meniča sa dosiahlo zníženie výkonu motora priemerne zo 7 192 W na 3 964 W, čím sa znížila spotreba elektrickej energie o 45 %. V peňažnom vyjadrení prišlo za jeden rok k úspore 40 680 Sk. Nadobúdacia cena meniča bola 65 000 Sk. Potom návratnosť investície je 1,6 roka a životnosť zariadenia sa predpokladá 8–10 rokov. Z údajov vyplýva, že využívanie meniča frekvencie v oblasti pohonu vývevy sa javí ako vysoko energeticky a ekonomicky efektívne. Pri využití meniča pred výkonnejším motorom (keď taký výveva požaduje) je možné dosiahnuť až 70% úsporu.

Kľúčové slová: dojacie zariadenie; výveva; frekvenčný menič; úspora energie

Corresponding author:

Doc. Ing. LUBOMÍR KUBINA, CSc., Slovenská poľnohospodárska univerzita, Mechanizačná fakulta, Trieda A. Hlinku 2, 949 76 Nitra, Slovenská republika
tel.: + 421 37 772 21 89, fax: + 421 37 741 70 03, e-mail: lubomir.kubina@uniag.sk

Analysis of tractor age and its development in Slovak Republic

V. KROČKO¹, R. MIKUŠ¹, R. DRLIČKA¹, F. ZACHARDA²

¹*Slovak University of Agriculture, Nitra, Slovak Republic*

²*Agriculture Technical and Testing Institute, Rovinka, Slovak Republic*

ABSTRACT: The paper deals with analysis of trends in state and wear degree of tractors in years 1995, 1997 and 1999. Basic information about numbers and age structure of tractors from randomly selected businesses from Trenčín country was obtained. Optimal indexes for description of machines ageing process and shabbiness are average age and wear degree. The development of tractors state and wear rate was computed by means of mathematic-statistical methods. To find out the trends of development regression analysis was used. Based on results of this work we can say that tractors in Slovak Republic are considerably worn and obsolescence has cumulative trend. Serviceability costs as well as operation costs of worn machines are respectable. Thus renovation intensity of machinery should have to improve as soon as possible.

Keywords: wear degree; obsolescence; lifetime; mathematic-statistical methods; tractors

The big structural and economical changes lately passed in agriculture. Trend in manpower decrease continues and must be replaced by machinery. In addition, existing machinery is highly worn in many cases hence it must be replaced by new and powerful machines. JECH (2000) states, that 88.2% of machines used in plant production in Slovak Republic were older than 8 years at beginning of year 2000. It confirms low level of machinery renovation. Thus it is necessary to accentuate the attention to machinery reproduction.

It follows that agricultural machinery obsolescence in Slovak Republic is very high. Therefore observations of machinery ageing process and obsolescence as well as trend in amounts of machines have a big value.

QUANTITATIVE STATE OF AGRICULTURAL MACHINERY

If we want to observe and analyse the development of agricultural machinery obsolescence, we need to find and analyse their quantitative state. Based on analysis made in agribusiness follows that numbers of machines are still declining (Table 1). This indicates the obsolescence of machinery. On the other side this declining is partially caused by purchases of new machines on higher technical level, reached higher seasonal efficiency, higher reliability and quality of work. This renovation is very slow due to shortage of money.

To determine an appropriate strategy of purchase and production of agriculture machinery we must first to find and analyse their quantitative state and trends in development. Therefore is needed to realize observations in some subsequent years to determine changes in age and performance machinery structure. It claims a systematic research in this area. Based on results of that research is possible to determine an appropriate strategy of agricultural machinery renovation (ZACHARDA et al. 1998).

OBSOLESCENCE OF AGRICULTURAL MACHINERY

Machines are subdued to continual quantitative and qualitative changes in production process. Aside they are worn thus they gradually lose their initial properties and they do not satisfy the production requirements. On the other side they are repaired, modernized or even replaced by a new machines.

ŠVANTNER (1980) defines wear of machines as a process during that machines transfer their value into the value of products. In this process machines lose progressive their utility. LAUČÍK (1974) defines wear as a progressive loss of inherent value of machines as well as utility of machines.

To find out the agricultural machines obsolescence is possible to use many indexes. Procedures that find out the machinery obsolescence directly by virtue of

Table 1. Development of tractors number in Slovak Republic in 1990–1999 (JECH 2000)

Type	1990	1991	1992	1993	1994	1995	1996	1997	1998	1999	
Tractors	Together	36,912	34,980	33,009	30,851	29,810	27,746	26,650	25,820	25,067	23,913
	New	1,405	172	109	132	122	233	419	500	514	110

diagnostic signals are most accurate but also most expensive. More easily and cheaper but also inaccurate are procedures based on operation time. This assumes relation between machine obsolescence and their operation time. From our point of view optimal indexes for description of machines ageing process and obsolescence are average age and wear degree. But these indexes not deliberate operation regimes and level of machines maintenance. To obtain basic information about situation in state and wear degree their accuracy is sufficient.

Wear degree (degree of physical depreciation) can be measured by many methods. Time method is used most frequently. It gives degree of physical depreciation by ratio of real age of machines to standard lifetime (ŠVANTNER 1980):

$$O_f = \frac{t_r}{t_n} \cdot 100 \quad (1)$$

where: O_f – degree of physical depreciation (%),
 t_r – real age of machine (year),
 t_n – standard lifetime of machine (year).

LAUČÍK (1974) designates this method as method of finding of physical depreciation according to lifetime. But it must be satisfied a condition of balanced wear development in relation to machines age. That wear degree value is informative only, because it does not deliberate operation conditions.

BLECHA (1974) denotes progressive machine parameter changes independent of operation conditions as ageing process.

It is possible to determine the wear degree from machines' work volume (LAUČÍK 1974):

$$O = \frac{T_f \cdot P_f}{V_n \cdot t_n} \cdot 100 \quad (2)$$

where: O – degree of physical depreciation (%),
 T_f – number of real worked years (year),
 P_f – average volume of real production,
 t_n – standard lifetime (year),
 V_n – annual production capacity or standard performance of machine.

The next method of wear degree defining is value expression by ratio of accumulated depreciation to acquisition prices of machines (ŠVANTNER 1980):

$$O_s = \frac{C_n - C_z}{C_n} \cdot 100 \quad (3)$$

where: O_s – machines wear degree (%),
 C_n – acquisition prices of machines (SKK),
 C_z – depreciated prices of machines (SKK).

The disadvantage of this method are zero depreciated prices after depreciation, i.e. wear achieves 100% and this value cannot be exceeded. Thus this method does not assume using of machines longer than depreciation time.

Real state of worn machine is possible to find out by machine inspection only, i.e. finding of wear degree by technical state (LAUČÍK 1974; ŠVANTNER 1980). This method is most accurate, but it claims a lot of time and practice.

ZACHARDA et al. (1998) proposed to express machines obsolescence by a coefficient called age wear degree. It is defined as ratio of average age (average operation time) to standard operation time (e.g. depreciation time):

$$MO = \frac{\bar{t}}{T_o} \cdot 100 \quad (4)$$

where: MO – age wear degree (%),
 \bar{t} – average age of machines (year),
 T_o – depreciation time (year).

After 100% age wear degree is reached we can consider the machine as "pensioner". It manages its work oftentimes, but by much lower performance and longer breaks (maintenance and repairs). If computed age wear degree for set of machines is more than 100% it signalsizes dominant portion of machines older than depreciation time. Thus the machinery is ancient.

MATERIALS AND METHODS

The aim of this work is to evaluate both the quantitative state and wear degree of tractors in years 1995, 1997 and 1999. The next problem is to find out and analyse trends in state and wear degree of tractors.

To assess current conditions of the agricultural machinery the data about age structure of tractors used in randomly selected agribusinesses from Trenčín county were used. All forms of businesses and production areas of Slovak agriculture were proportionally represented in the selection.

Tractors were into 4 age categories arranged (under 4 years, 4–8 years, 8–12 years, more than 12 years).

1. Firstly we computed absolute rate values for individual age categories ($n = 4$) for all businesses ($N = 88$):

$$m_i = \sum_{k=1}^N m_{ki}, \text{ for } i = 1, \dots, n \quad (5)$$

where: m_i – absolute rate of i^{th} age category,
 m_{ki} – absolute rate of i^{th} age category of k^{th} business.

2. From absolute rate values we computed the relative rate values of i^{th} age category of tractors:

$$p_i = \frac{m_i}{\sum_{i=1}^n m_i} \quad (6)$$

where: $\sum_{i=1}^n m_i$ – general number of tractors

and cumulative relative rate values of i^{th} age category of tractors:

$$P_i = p_i + p_{i-1} \quad (7)$$

where: $p_{i-1} = 0$ for $i = 1$.

3. Values computed from equations (5), (6) and (7) are written into Table 2 and graphically displayed by histogram and empirical distribution function curve.

4. We computed the average value of tractors age:

$$\bar{t} = \sum_{i=1}^n t_{avr} \cdot p_i \quad (8)$$

Table 2. Quantification of evaluated parameters for tractors (KROČKO et al. 2000)

Type of machine		Tractors								
Year	1995			1997			1999			
Index	m_i	p_i (%)	P_i (%)	m_i	p_i (%)	P_i (%)	m_i	p_i (%)	P_i (%)	
Age category	under 4 yrs	47	2.55	2.55	80	4.15	4.15	78	4.16	4.16
	4–8 yrs	287	15.59	18.14	134	6.95	11.10	67	6.57	7.73
	8–12 yrs	579	31.45	49.59	601	31.17	42.27	435	23.18	30.90
	over 12 yrs	928	50.41	100	1,113	57.73	100	1,297	69.10	100
	together	1,841			1,928			1,877		
Average age (year)		11.19			11.70			12.29		
Standard deviation (year)		6.96			6.71			6.26		
Age wear coefficient		1.40			1.46			1.54		

where: t_{avr} – centre of i^{th} age category (year),

p_{ij} – relative rate of tractors of i^{th} age category,

5. and the standard deviation of tractors age:

$$s = \sqrt{\sum_{i=1}^k (t_i - \bar{t})^2 \cdot p_i} \quad (9)$$

6. Finally we computed age wear coefficient:

$$KO = \frac{\bar{t}}{T_{oj}} \quad (10)$$

where: KO – age wear coefficient,

\bar{t} – average age of tractors (year),

T_{oj} – depreciation time (year).

RESULTS AND DISCUSSION

Basic information about numbers and age structure of tractors was obtained from randomly selected businesses from Trenčín country at years 1995, 1997 and 1999. This sampling set enable to predict the development of

age structure and wear rate of tractors in whole Slovak Republic.

Original data and computed indexes for years 1995, 1997, 1999 are listed in Table 2. The table is appended by both histogram and empirical distribution function curve of tractor distribution into individual age categories for individual years of observation (Fig. 1). In Fig. 2 graphical presentation of age wear coefficient development for individual years of observation is expressed.

ANALYSIS OF ACHIEVED RESULTS

From Table 1 is clear that the increment of tractors rate in age category below 4 years, in percentage was from 2.55% in 1995 to 4.16% in 1999. But the increment of tractors older 8 years is much faster, from 81.86% in 1995 to 92.28% in 1999. These data are similar to that shown by JECH (2000). Variation between these data is around 5%, variation between trends of de-

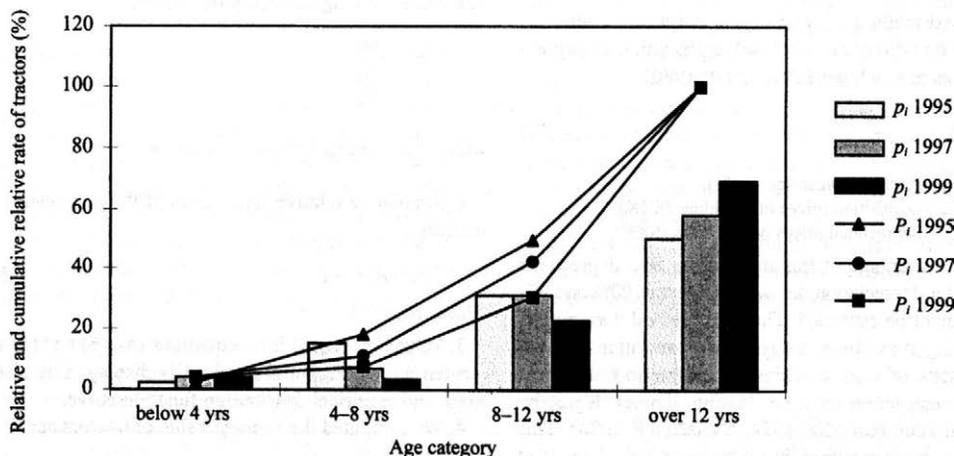


Fig. 1. Number of tractors in relation to their age

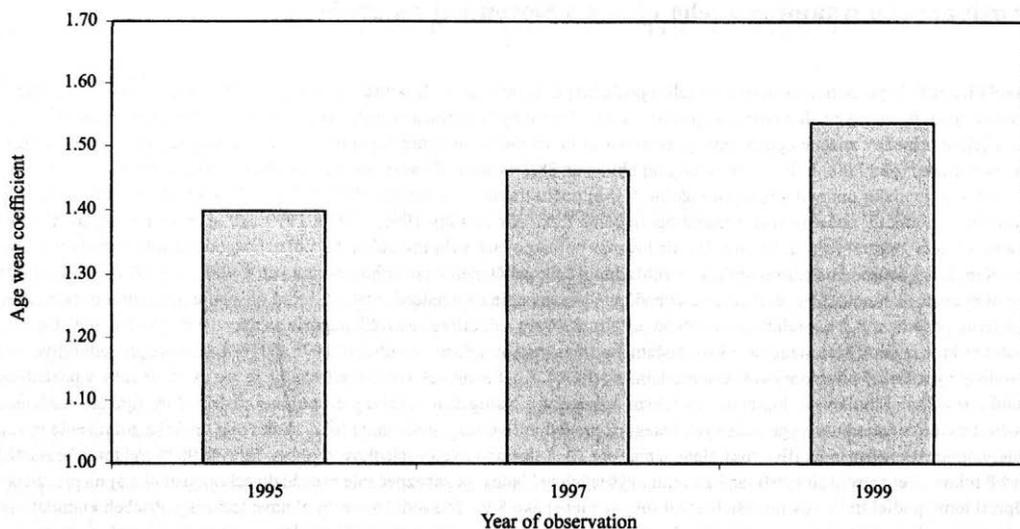


Fig. 2. Graphical evaluation of age wear level of tractors

velopment is less than 1%. Average age of tractors has cumulative trend too, from value 11.19 years in 1995 to value 12.29 years in 1999. Age wear coefficient increases from 1.40 in 1995 to 1.54 in 1999.

From Fig. 1 result soft increase of tractors number in below 4 years age category. In next two age categories this trend is opposite. It is clear to see tractors rate decrease in these age categories. In 4–8 years age category this decreasing is very large (from 15.59% to 6.95%) in years 1995–1997 particularly. Similar trend is in 8–12 years age category. Tractors from that age category are exploited even though their lifetime had passed. Alarming state is in over 12 years age category. Its rate is increased from 50.41% in 1995 through 57.73% in 1997 to 69.10% in 1999, i.e. in this age category occur nearly $\frac{3}{4}$ of all tractors.

Fig. 1 shows that cumulative relative rate curve is convex. It is caused by relatively little rates of age categories below 4 years, 4–8 years, 8–12 years and greater rate of over 12 years age category.

With decrease of tractor numbers in individual age categories, the convexity of curve increases (curve deepen). In 1999 the cumulative relative rates of tractors show some variances compared to 1995. While in 1995 the value of cumulative relative rate of tractors younger 8 years was 18.14%, in 1999 it was only 7.73%. Rate of undepreciated tractors is thus less than 10%.

Fig. 2 shows that age wear coefficient of tractors increases from 1.40 in 1995 to 1.54 in 1999, thus almost about 14%.

CONCLUSION

From sampling file of agribusinesses from Trenčín country data about both numerous state and age structure

of tractors were obtained. Based on that information we can forecast situation in all Slovak Republic with a particular probability.

Achieved results follow that more 90% of tractors are older than 8 years. These machines are used at the cost of escalating operational costs to secure their operational capability. Compared to the rate of tractors younger than 4 years is less than 5%. It indicates a little renovation of machinery.

Cumulative relative rate curve of tractors is convex, but it tends to deep. It indicates a decrease of tractors number in age of 4–12 years.

Established age wear coefficient shows continually increasing of tractors obsolescence, from value 1.40 in 1995 to value 1.54 in 1999. To more accurate assessment of development trends it is necessary to observe given file in next years.

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Analýza veku traktorov a jeho vývoja v Slovenskej republike

ABSTRAKT: V poľnohospodárstve nastali v poslednej dobe rozsiahle štrukturálne a ekonomické zmeny. Naďalej sa udržuje trend úbytku pracovných síl z poľnohospodárstva, ktorý musí byť nahradzaný technikou. Navyše, keďže existujúca technika je vo väčšine prípadov značne opotrebená, je potrebné aj ju nahrádzať novými, často výkonnejšími strojmi. JECH (2000) uvádza, že na začiatku roku 2000 bolo v Slovenskej republike až 88,2 % strojov používaných v rastlinnej výrobe starších ako 8 rokov, čo potvrdzuje nízku obnovu strojového parku. Vývoj počtu traktorov v rokoch 1990–1999 v SR je uvedený v tab. 1. Cieľom práce je vyhodnotiť početný stav a mieru opotrebenia traktorov za roky 1995, 1997 a 1999 ako aj zistiť a analyzovať vývoj stavu a miery opotrebenia traktorov. Do sledovania bol zapojený vybraný súbor 88 poľnohospodárskych podnikov z kraja Trenčín, ktorý svojou štruktúrou podnikov zohľadňuje celkové členenie poľnohospodárskych podnikov v SR ako aj charakter výrobných podmienok. Zdrojové údaje a vypočítané ukazovatele sú uvedené v tab. 2, ktorá obsahuje absolútnu početnosť m , relatívnu početnosť p , a kumulatívnu relatívnu početnosť P , pre jednotlivé vekové kategórie za roky 1995, 1997 a 1999. Celkový počet traktorov umožňuje sledovať vývoj početného stavu strojov v časovom období 1995–1999. Takisto je pre jednotlivé roky uvedený vypočítaný priemerný vek a smerodajná odchýlka. Koeficient vekového opotrebenia je pre uvedené roky v poslednom riadku tabuľky. Tabuľky sú doplnené grafickým zobrazením histogramu a krivky empirickej distribučnej funkcie rozdelenia počtu traktorov do jednotlivých vekových kategórií pre jednotlivé roky sledovania (obr. 1) ako aj o grafické zobrazenie vývoja miery opotrebenia pre jednotlivé roky sledovania (obr. 2). Z dosiahnutých výsledkov vyplýva, že vyše 90 % traktorov je starších než 8 rokov. Tieto stroje sú využívané za cenu zvýšených nákladov na zabezpečenie prevádzkyschopnosti ako aj na prevádzku. Oproti tomu podiel traktorov mladších než 4 roky je menej ako 5 %. To svedčí o nízkej obnove techniky. Priebeh kumulatívnej relatívnej početnosti traktorov je konvexný, pričom má tendenciu prehlbovať sa. To svedčí o poklese počtu traktorov vo veku 4–12 rokov, ktorý je sprevádzaný výrazným nárastom počtu strojov vo veku nad 12 rokov. Nami zavedený koeficient vekového opotrebenia taktiež ukazuje na pokračujúce zväčšovanie opotrebovanosti traktorov z hodnoty 1,40 v roku 1995 na hodnotu 1,54 v roku 1999. Na presnejšie stanovenie trendov vývoja je potrebné sledovanie daného súboru v ďalších rokoch.

Kľúčové slová: miera opotrebenia; opotrebovanosť; životnosť; matematicko-štatistické metódy; traktory

Corresponding author:

Doc. Ing. VLADIMÍR KROČKO, CSc., Slovenská poľnohospodárska univerzita, Mechanizačná fakulta, Trieda A. Hlinku 2, 949 76 Nitra, Slovenská republika
tel.: + 421 37 772 21 87, fax: + 421 37 741 70 03, e-mail: vladimir.krocko@uniag.sk

Determining of the power requirements of disk mower with conditioner

J. ŠESTÁK, J. PRŠAN, CS. GYURIÁN

Slovak Agricultural University, Faculty of Agricultural Engineering, Nitra, Slovak Republic

ABSTRACT: To determine the power requirements of the disk mower aggregate with in-line mounted drum trimmer (conditioner) and separately the power requirements of conditioner, field experiments were carried out. These experiments were implemented for several combinations of technological parameter set-ups of the conditioner. The mechanical effect of the conditioner was increased by changing the gap between the rotor fingers of the conditioner and its cover by closing the shield, as well as creating obstacle in the fodder flow, by a comb. The power parameters were investigated in the most available fodders, the grass-mixture and lucerne. The highest power requirements of the aggregate were found through the harvest and mechanical treatment of grass, at the value 21.2 kW, when the feeding was 11.71 kg/s. Through the harvest and mechanical treatment of lucerne the power requirements of 18.48 kW at the throughput of 10.39 kg/s were found. The power requirements of the conditioner according to the feed, do not rise equidistantly with the power requirements of the aggregate, as it was specified after evaluating, too. The technical-exploitation parameters of the aggregate will be:

- The specific power requirements of the aggregate:
 - for the harvest and treatment of grass, 10 kW/m (of working width),
 - for the harvest and treatment of lucerne, 8.97 kW/m (of working width).
- The specific power requirements of the conditioner:
 - through the harvest and treatment of grass, 8.2 kW/m (of working width),
 - through the harvest and treatment of lucerne, 6.8 kW/m (of working width).

These specific power requirements are defined at the fodder capacity of ~ 10 kg/s.

The specific power in the no-load run:

- for the whole aggregate, 3 kW/m (of working width),
- for the drum trimmer, 1 kW/m (of working width).

Keywords: harvest of fodders; disk mower; conditioner; power requirements

To secure a fast harvest, with increased proportion of solids and with the minimal loss of nutrients, it is suitable together with the mowing to treat the fodder. To do this, several machine types of mechanical treatment can be used, KOROLOVICS (1979), KLINER (1975). A significant exploitation parameter, determining the type of driving machine, is the total power input of the technologic units, the mower and the conditioner. Power requirements of mowers with grinder were approached by ČERMÁK and HORA (1969). SOUČEK (1986) presented a conceptual solution and energy requirements evaluation of rotational mowers with treating machines. Nonlinear dependency of power requirements on mass flow was found in every observed machine.

SEDLÁK and BAUER (1999) experimentally determined the energy requirements of disk mower ŽTR-285D with working width of 2.85 m. For the aggregate assembled from a disk mower and in-line mounted conditioner, the total power requirements changes were defined nonlinearly, on the other hand the power requirements of the conditioner linearly.

The results of power requirements measurements of aggregate assembled from disk mower and conditioner,

and the power to drive the conditioner, are presented in this article. Grass and lucerne were treated in the field tests.

MATERIAL AND METHODS

The mower MARAGON MD 5 assembled with a drum trimmer carried out the field tests. The overall view of the aggregate can be seen in Fig. 1; the basic technical and exploitation parameters are in Table 1.

The experiments were carried out in the location of Žirany, part of the School agricultural farm Koliňany, through June 20 to June 28, 1999.

The terrain parameters can be seen in Table 2.

The humidity of soil and plants was determined in accordance with the Standard STN 46 7092. The penetration resistance was measured in accordance with the STN 47 0121.

The input characteristics of plants are described in Table 3.

The methodical order of measurement of power requirements changes of the whole aggregate, as well as of the conditioner, were carried out in accordance with the scheme shown in Fig. 2.

Table 1. Basic technical and exploitation parameters of the aggregate

Tractor	Z-12011
PTO speed (rpm)	540
Working speed (m/s)	0.68–2.4
Working width of the mower (mm)	2,050
Type of the mower	MARAGON MD 5
Number of working discs (pcs)	5
Number of knives per disc (pcs)	2
Width of the trimmer rotor (mm)	1,400
Diameter of the trimmer rotor (mm)	550
Number of fingers on the rotor (pcs)	18
Rotor speed (rpm)	750
Positioning of the aggregate	Hydraulically changed

Table 2. Terrain parameters

Tilt angle of the pavement (deg.)	3–5
Density of the soil (kg/m ³)	1,430
Moisture content in the soil (%)	40.80
Penetration resistance of the soil (MPa)	4

Table 3. Stand characteristics

Parameter	Grass	Lucerne
Mean yield (kg/m ²)	2.51	2.25
Mean height of the plants (m)	0.76	0.48
Moisture content (%)	28	32

For the set-up of technological position, respecting the shield and comb position, then corresponding five values of working speed (resp. feeding). Sheet doors, inserted into the mass flow were marked as shield, and a component with fingers adjusted into the mass flow was marked as comb. These are the parts of the technical documentation of adapter.

The feed of fodder mass through the machine was determined by the weight of yield per m² (harvest), the mean working width of disk mower and the working speed of machine, with the use of formula:

$$q = m \cdot B \cdot v \quad (1)$$

where: m – the yield of crop (kg/m²),

B – the working width of disk mower (m),

v – the working speed of machine (m/s).

The working speed (mean speed at the selected gear level) was calculated from the measured time, whilst the aggregate treated a 20m long parcel.

The power requirements was specified from the measured torque and angular velocity. Both parameters were measured on the p.t.o. shaft of the tractor (to define the total power input), and on the driving shaft of the conditioner (to define the power required to drive the conditioner).

The disposition of torque and angular velocity sensors is shown in Figs. 3 and 4.

The torque in the p.t.o. shaft of the tractor was measured by dynamometer Hottinger-Baldwin MD-355, which has the capability to measure up to 1,000 N · m. By calibration process we found out the modulus of this dynamometer. This has the value:

$$\mu_H = 420 \text{ N} \cdot \text{m/V}$$

where: V – the output calibration voltage.

The torque in the drive shaft of the conditioner was measured by a dynamometer made by fa VZLU-Letňany, with the capability to measure up to 1,200 N · m. By calibration we found out the modulus:

$$\mu_K = 150 \text{ N} \cdot \text{m/V}$$

The angular velocity measurements on the p.t.o. shaft of the tractor as well as on conditioner drive shaft were carried out by tachogenerators, made by MEZ Náchod, type K4A2. We specified the modulus of rotation frequency by its calibration:

$$\mu_{(\omega)} = 19.5 \text{ (rad/s)/V}$$

The gathered values of torques and angular velocity on the p.t.o. shaft of the tractor as well as on the conditioner drive shaft, for the selected kinematical and technological parameters, are stored on the tape-recorder Kraus



Fig. 1. The overall view of the aggregate

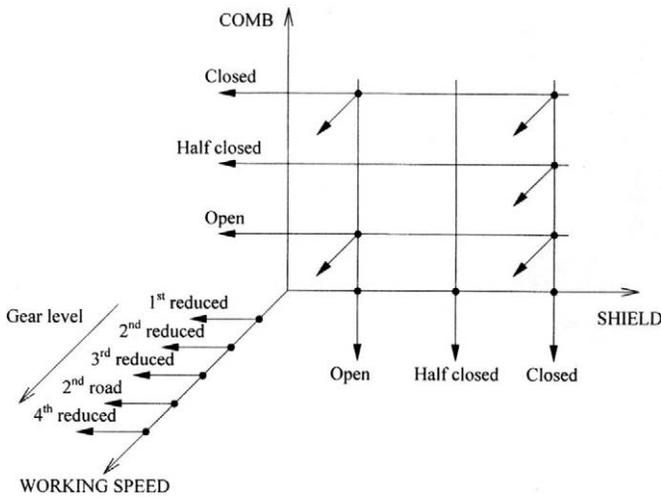


Fig. 2. The measurement method

Messtechnik (KMT), type COMP 32. The sampling speed of 120 kbit/s was used.

The control tests of recorded values were carried out on a coordinate plotter BAK 5T, with the sensitivity of 100 mV/cm.

Global view of the measurement disposition can be seen in Fig. 5.

The recorded analog signals of voltages, representing the values of torque and angular velocity, were treated according to the scheme in Fig. 6.

The A/D converter, made by firm BCM system GmbH, type N-Meter 4, had these parameters:

- number of channels: 4,
- range of input voltage: 0 ÷ 10V,
- drift: ± 50 mV.

The files of discrete representation of experiments, with contemporary filtration of high frequency components, were stored in PC. The sampling time was 0.004 s.

The mean value of torque, for one capacity and technological set-up of the conditioner, can be defined by this formula:

$$M_k = \frac{\sum_{i=1}^{i=n} M_{k(i)} \cdot \Delta t}{t} \quad (2)$$

where: $M_{k(i)}$ – the value of torque in time step i (N · m),
 Δt – the time step (s),
 t – the duration of experiment (s).

The kinematical compatibility of rotation frequency with the torque was evaluated from its discrete representation. Then the mean value of angular velocity will be:

$$\omega = \frac{\sum_{i=1}^{i=n} \omega_{(i)} \cdot \Delta t}{t} \quad (3)$$

where: $\omega_{(i)}$ – the value of angular velocity in time step i (rad/s).

Then for the defined kinematical and technological set-up of the aggregate and conditioner, the sought mean value of the power requirements was specified:

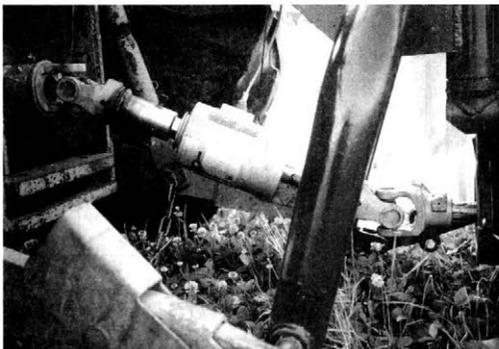


Fig. 3. The disposition of torque and rotation frequency sensors on the main cardan



Fig. 4. The disposition of torque and rotation frequency sensors on the conditioner cardan

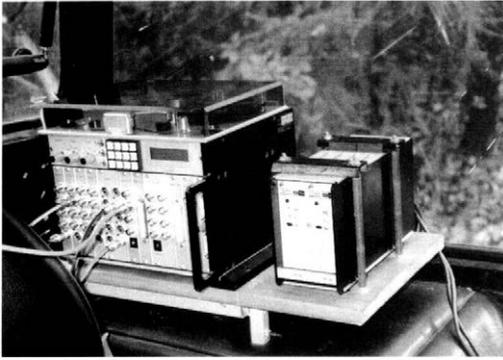


Fig. 5. Global view of the measurement disposition

$$P(D) = M_k(D) \cdot \omega(D) \quad (4)$$

where: (D) – symbolizes the mean value of power requirements, torque and angular velocity.

The original records of torque and angular velocity behaviors of the main cardan and the cardan of conditioner are shown in Figs. 7 and 8. The pull-up phase of angular velocity of both shafts is moved, because the no-load run is recorded with the “complete” angular velocities of shafts. From the prompted graphs the harmonic (deterministic) component is viewable, caused by the system of serially arranged Hook’s joints in cardan shafts. The high demonstrativeness of the effect of harmonic components, caused by cardan mechanisms, is clearly shown in behaviors of torques and angular velocities.

The first from the evaluation of power requirements, in the technological operation of mowing and mechanical treatment of fodder, is the combination of limit technological set-ups, when the shield and the comb are fully closed. The dependency of power requirements on the feeding mass was imprinted in Fig. 9. In the graphical course first of all the power requirements in no-load run for the main cardan with the value of 5.93 kW, and

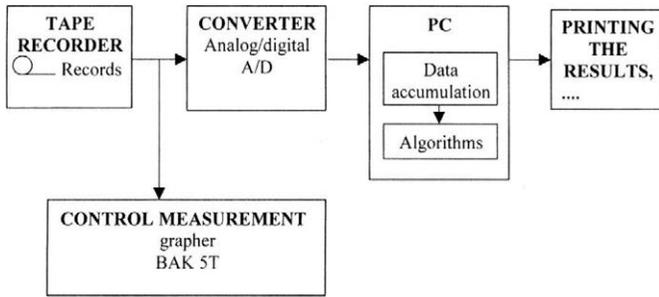


Fig. 6. The scheme of the data treatment

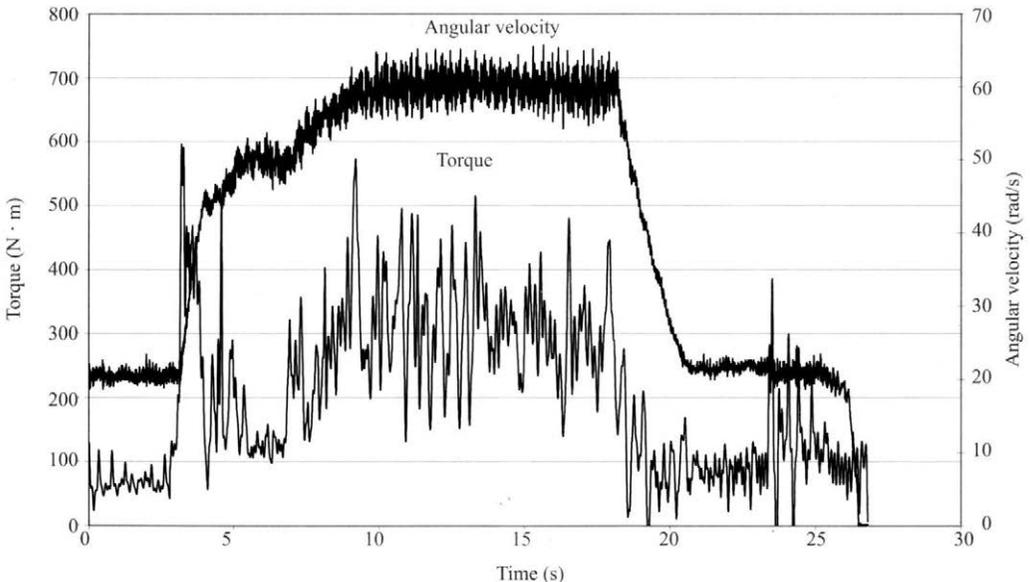


Fig. 7. The original records on the p.t.o. shaft

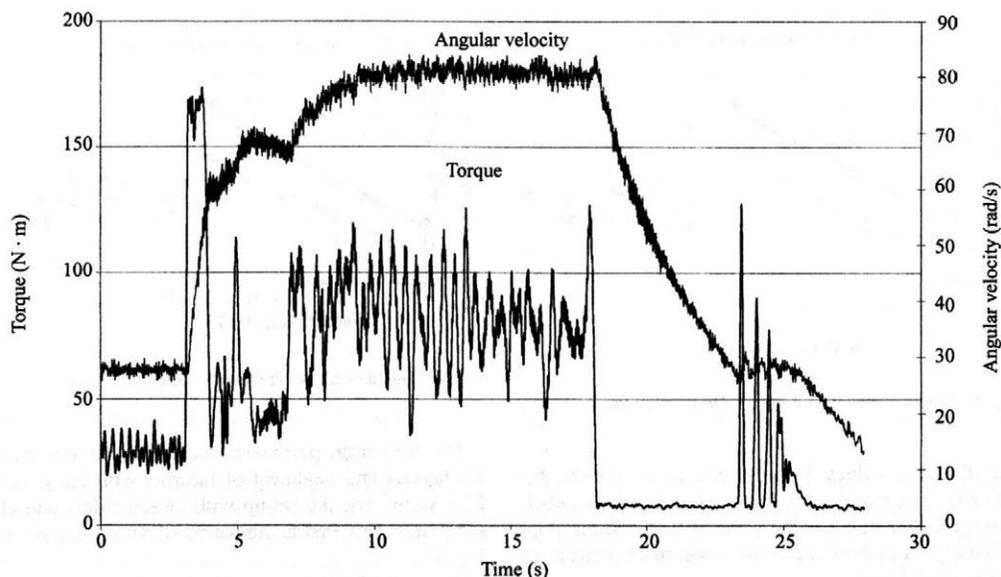


Fig. 8. The original records of the conditioner shaft

for the conditioner with the value of 1.903 kW, were shown. A linear approximation from the measured values, in both cases, after the tests, with the correlation coefficient of $k = 0.9658$ for the main cardan and $k = 0.9795$ for the cardan of conditioner, proved to be significant (statistically). The fact, that the power requirements to drive the conditioner changes analogically as the power requirements of the whole aggregate is significant, however, its intensity at the lower values of capacity decreases. The evidence of this fact serves the comparison of the slopes of approximation lines, when the whole power requirements has the slope of 1.323 and the conditioner has 1.21.

The second peripheral technological set-up at the harvest of grass was to fully open the shield and the comb on the drum trimmer. Though, this technological set-up does not represent 'zero' influence on the treated fodder.

The flow of the treated fodder is scraped and compressed by the inserted shield and at the same time 'stalked' by the comb (at least on the sides of the mass flow).

The dependency of the power consumed by the whole aggregate, as well as by the conditioner on the feeding, is drafted in Fig. 10. The fact, that by increasing the capacity the power to drive the conditioner does not rise identically with the total power requirements. The slope of the approximation line of the total power requirements, which has the value of 0.945, is lower than the slope of the approximation line, representing the power to drive conditioner with the value of 0.9643.

The first from the combination of mid-range technological set-ups was the set-up with closed shield and open comb of drum trimmer. This is drawn in Fig. 11. The power course to drive the conditioner towards the power requirements course of the whole aggregate in

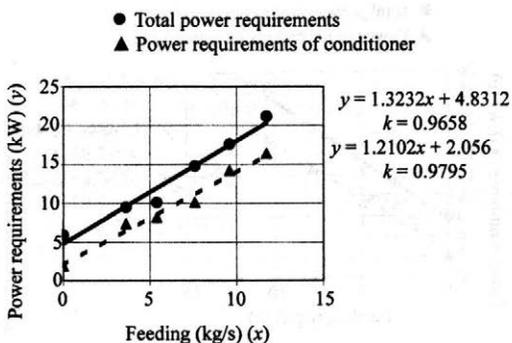


Fig. 9. Shield closed, comb closed – Grass treatment

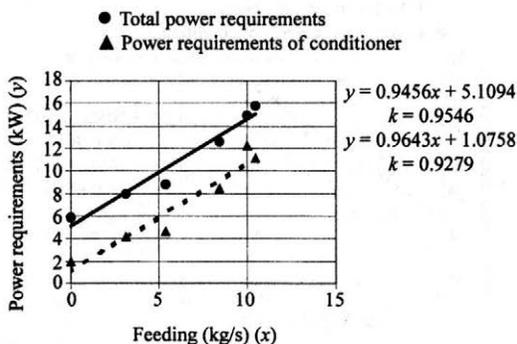


Fig. 10. Shield open, comb open – Grass treatment

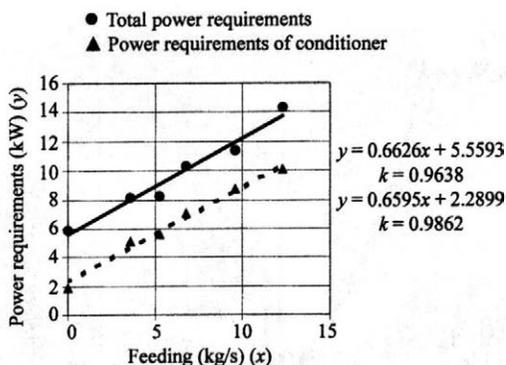


Fig. 11. Shield closed, comb open – Grass treatment

the efficiency values does not change identically, because the slope of the approximation line of the whole power requirements has the value of 0.66 and the slope of the approximation line of the power to drive the drum trimmer has 0.659.

The second combination of mid-range technological set-ups was a case when the shield was fully open and the comb completely closed. The measured and evaluated values are shown in Fig. 12.

The comparison between influences of single mechanical components, the closed shield and closed comb is significant, too. Comparing the courses drawn in Figs. 11 and 12 we found out that the total power to drive the aggregate with closed comb at almost the same feeding (for the closed shield 12.3 kg/s and the closed comb 11.7 kg/s) is by 20% higher than with the closed shield. The drag effects of the comb, caused by 'hang up' and stretching the fodder through the gaps between fingers are higher than the 'abrasion' influence of the shield with smooth sheet.

The last technological set-up was the combination of closed shield and half closed comb. This is drawn in Fig. 13.

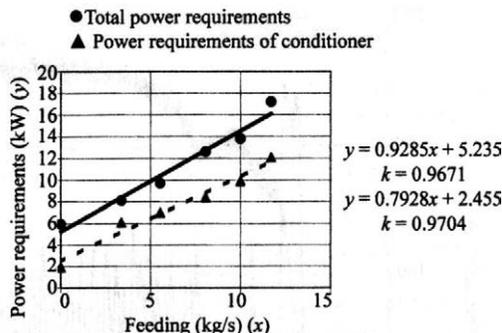


Fig. 12. Shield open, comb closed – Grass treatment

The first from peripheral technological set-ups, in the harvest and treatment of lucerne, with the yield of 2.25 kg/m² was the set-up with closed shield and closed comb. This first combination of set-up is shown in Fig. 14.

The mutual comparison of the power requirements for grass, Fig. 9, and lucerne, Fig. 14, at closed shield and fully closed comb indicates that this requirements is relatively higher for the harvest and treatment of grass than for lucerne. In the observed range of feeding changes (with high possibility) this is caused by the higher crop yield of grass (2.51 kg/m²) than lucerne (2.25 kg/m²). This matches the power requirements accretion, which depends on the capacity, too.

The second peripheral technological set-up at the harvest and treatment of lucerne, was fully open the shield and close the comb. These changes of the total power requirements of aggregate and conditioner are drawn in Fig. 15. In comparison with the same set-up for grass, we found out that the total power requirements at the harvest of lucerne is relatively lower than at the grass harvest.

The first intermediate set-up of the shield and comb was the set-up of the shield as fully closed and the comb fully open. This course of the power requirements is

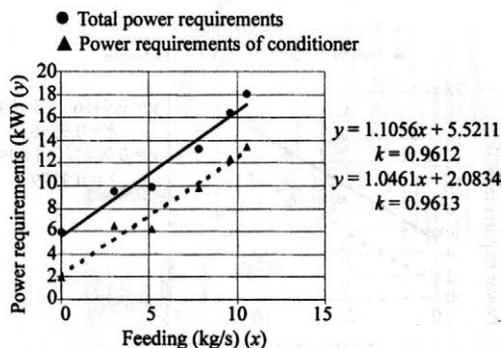


Fig. 13. Shield closed, comb half closed – Grass treatment

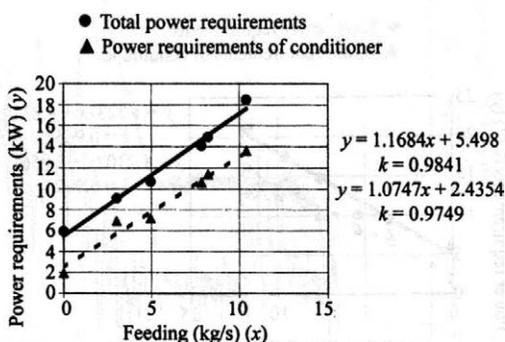


Fig. 14. Shield closed, comb closed – Lucerne treatment

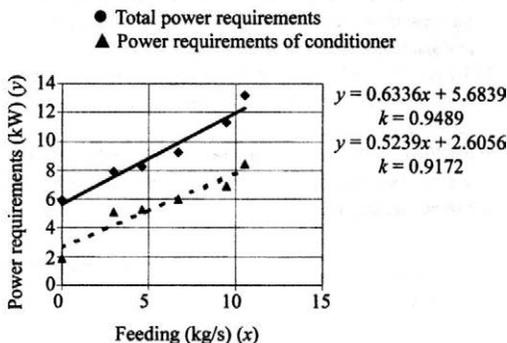


Fig. 15. Shield open, comb open – Lucerne treatment

drawn in Fig. 16. Linear regression with parameters shown in this figure, proved to be statistically evident. Parallel comparison of power requirements between grass, Fig. 11, and lucerne, Fig. 16, indicated that higher requirements shows the harvest of grass compared to the lucerne. Analogically the velocity of power requirements accretion is for the 'almost identical' capacities higher with grass than with lucerne.

The second mid-range set-up and its power requirements, when the shield was fully open and the comb completely closed, is drawn in Fig. 17. Linear approximation, in the interval of the measured capacities of mass, what is marked in the figure, too, was statistically confirmed. The comparison of power requirements for the fully closed shield (Fig. 16) and closed comb (Fig. 17) shows that this input is higher for the closed comb.

Simultaneous comparison of the power requirements at the harvest of grass (Fig. 12) and lucerne (Fig. 17) proves that the input is higher at the grass harvest. The assumption of the higher drag effects of grass in the finger gaps of the comb, compared to the effects of shield, proved to be valid.

The last mid-range technological set-up is the fully open shield and half closed comb. This is in Fig. 18. Linear approximation of measured values is statistically documented in this figure.

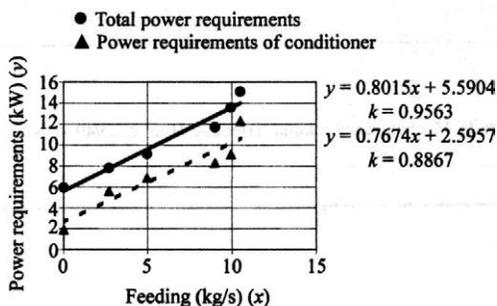


Fig. 17. Shield open, comb closed – Lucerne treatment

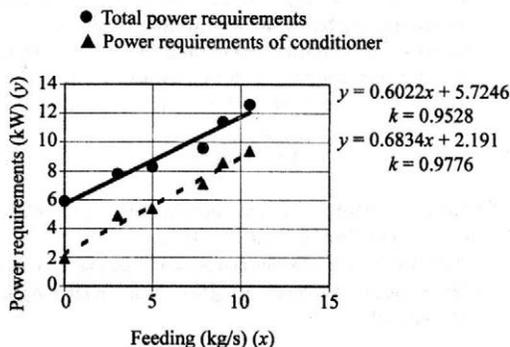


Fig. 16. Shield closed, comb open – Lucerne treatment

By summarizing the power requirements for the harvest and treatment of grass and lucerne we found:

- the total power requirements of the aggregate (mower-conditioner), depends on the feeding the machine, changes linearly. Our linearized approximations is corresponding to the results carried out by KLINNER (1975) and SEDLÁK and BAUER (1999). The aggregate is showing the highest power requirements at the harvest and treatment of grass with the value of 21.2 kW, at the feeding of 11.719 kg/s; at the harvest and treatment of lucerne 18.48 kW, at capacity of 10.39 kg/s. The power to drive the conditioner at the mechanical treatment of grass, at the feeding of 10.71 kg/s, was 16.4 kW, and for lucerne 13.61 kW with the feeding of 10.39 kg/s. The listed values match the technological set-up, when the shield and the comb are fully closed. For the technological-exploitation valuation of the machine the following values are listed:

The mean power requirements of the aggregate mower-conditioner (with five disks) per one meter of working width and feeding of 10 kg/s is:

- 10 kW/m – for grass harvest,
- 8.97 kW/m – for lucerne harvest,
- power requirements in no-load run of the whole aggregate for one meter of working width – 3 kW/m,

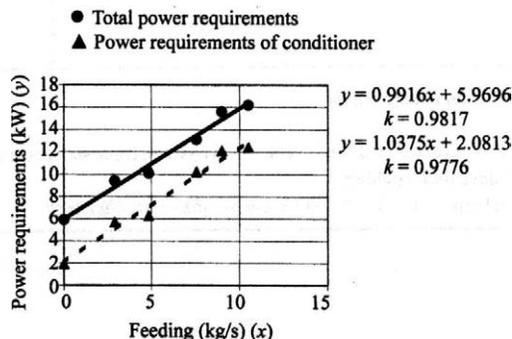


Fig. 18. Shield closed, comb half closed – Lucerne treatment

- power requirements in no-load run of the drum trimmer for one meter of working width – 1 kW/m,
- the power requirements accretion to drive the drum trimmer (conditioner) is not equivalent to the power requirements accretion of the whole aggregate.

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Stanovenie energetickej náročnosti diskovej kosačky s kondicionérom

ABSTRAKT: Uskutočnili sme poľné experimenty prototypu agregátu diskovej kosačky so sériovo zaradeným lámacím bubnom s cieľom stanoviť potrebu príkonu celého agregátu a potrebu príkonu lámacieho bubna. Pokusy sme vykonali pre rôzne kombinácie nastavenia technologických parametrov kondicionéra. Mechanický účinok kondicionéra sme zvyšovali zmenou medzery medzi prstami rotora kondicionéra a jeho plášťom, privieraním plechovej clony ako aj vytváraním prekážky v toku krmoviny pomocou prečesávacieho hrebeňa. Príkonové exploatačné parametre sme zisťovali u v praxi najzastupiteľnejších krmovínach, a to trávinatej zmesi a lucerny. Najvyššiu hodnotu potreby príkonu mal agregát pri zbere a mechanickom ošetrovaní trávinatej hmoty, a to 21,2 kW, keď priechodnosť hmoty bola 11,71 kg/s. Pri zbere a ošetrovaní lucerny sme zistili najvyššiu potrebu príkonu v hodnote 18,48 kW, a to pri priechodnosti hmoty 10,39 kg/s.

Vyhodnotením sme tiež stanovili, že príkon na kondicionér so zvyšovaním priechodnosti nestúpa ekvidištantne s príkonom agregátu. Technicko-exploatačné parametre agregátu budú:

- merný príkon celého agregátu:
 - pre zber a ošetrovanie trávy, 10 kW/m (záberu),
 - pre zber a ošetrovanie lucerny, 8,97 kW/m (záberu),
- merný príkon kondicionéra:
 - pre zber a ošetrovanie trávy, 8,2 kW/m (záberu),
 - pre zber a ošetrovanie lucerny, 6,8 kW/m (záberu).

Všetky uvedené merné príkony sú definované pre priechodnosť obidvoch krmovín v hodnote ~ 10 kg/s.

Merné príkony pre beh naprázdno:

- pre celý agregát, 3 kW/m (záberu),
- pre lámací bubon, 1 kW/m (záberu).

Kľúčové slová: zber krmovín; disková kosačka; kondicionér; výkonová potreba

Corresponding author:

Prof. Ing. JOZEF ŠESTÁK, CSc., Slovenská poľnohospodárska univerzita, Mechanizačná fakulta, Trieda A. Hlinku 2, 949 76 Nitra, Slovenská republika

tel./fax: + 421 377 33 60 73, e-mail: jozo.sestak@post.sk

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