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FLOW CHARACTERISTICS OF THE DECONTAMINATION PROCESS FOR NUCLEAR POWER STATIONS USING DEKOZ-PG I INSTALLATION

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Abstract

The DEKOZ-PG I apparatus is utilised for the circulation of decontaminant fluids. It is equipped with a propeller pump with unidirectional operation.

An assessment of the process technology concluded that the flow direction of the decontaminant fluids should be reversible by simply switching the drive motor revolutions between clockwise and anti-clockwise modes. This should result in a uniform cleansing of the steam generator walls.

In order to fulfil the above requirement, a new impeller had to be designed. The new design was based on information relating to the existing data: n (pump speed), Q (volume flow rate) and H (delivery head) as well as on measurements taken during a visit to the actual site.

The new design had to be based on an impeller system with blades that are symmetrical both in axial and in cross direction. This should ensure the proper functioning of the unit irrespective of the direction of drive motor rotation.

Keywords: decontamination, nuclear power station, propeller pump, symmetrical blades.

1. Aim

The function of the DEKOZ-PG installation is to decontaminate steam generators in nuclear power stations by chemically removing radiation.

The DEKOZ-PG I unit examined here is a column system comprising a propeller pump fitted with inlet (suction) and outlet (pressure) pipes and installed in the hot end head (‘collector’) of the heat exchanger.

Its function is the continuous circulation of a hot aqueous solution through the heat exchanger and the cold end head (‘collector’). If the pump moves the liquid at the planned rate, the whole system is then rinsed through in one minute at an approximate volume of 10 m^3. Using the prevailing method, the decontamination is carried out in a number of stages and it lasts several hours.

Nevertheless, according to measurements the efficiency of the method is less than satisfactory, *Fig. 1* demonstrates the extent of contamination (mGy/h) before and after the clean up process. The amount of undesirable residual contamination (as indicated by the grid-line area) points to a lack of uniformity in the cleaning process. This is also demonstrated here by the efficiency curve (*Fig. 1* – ‘DF’);

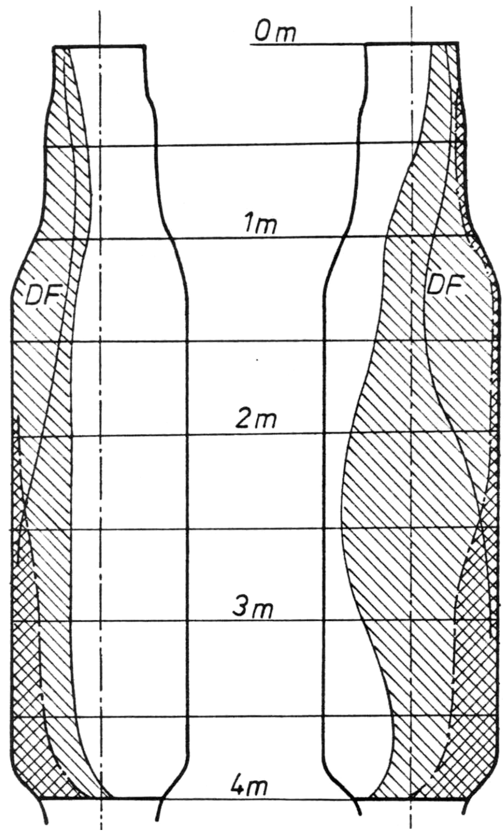

Fig. 1. Extent of contamination (mGy/h) in the hot end and cold end heads before (lined area) and after (grid-lined area) the chemical decontamination process

the upper section in both the hot end and cold end heads ('collector') are fully decontaminated, while the lower sections are cleaned in part only.

The evaluation of the process involved the study of the assembly drawing of the full heat exchanger, the installation drawing and description of the DEKOZ-PG I apparatus. It was also possible to visit the site and to make further measurements there. Since only the major dimensions were presented on the available drawings, a sketch had to be prepared of the opening on the suction and pressure sides of the pump (*Fig. 2*). The characteristic curve for the pump was not available and it is not possible to compute the flow resistance parameters in the system.

However, the above obstacles did not thwart the completion of this project and the formulation of conclusions and recommendations (See Clause 5). Nevertheless, they made it necessary to round off some of the values in the calculation.

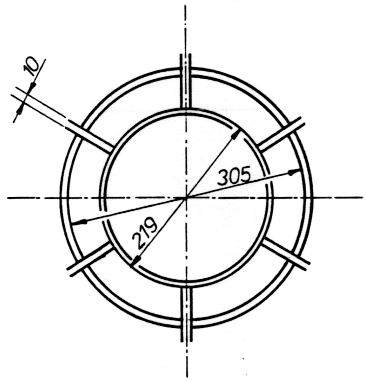

Fig. 2. Ribbed ring opening on the pressure and suction tail-pipes of the pump

Table 1. Cross sectional dimensions as per *Fig. 3*

Description	Diameter		Surface	
	O.D.	I.D.	Designation	Area
	mm			m^2
Impeller	300	130	A_0	0.0573
Ribbed ring (*Fig. 2*)	305	219	A_{bgy}	0.0301
Outlet (pressure side) pipe	200	55	A_{ny}	0.0290
Inlet (suction side) pipe	200	–	A_{sz}	0.0314
1 hole each on inlet and outlet (suction and pressure side) pipes	30	–	A_1	0.0007
1 opening on inlet at the bottom	155×70		A_t	0.0108

Table 2. Average flow velocities as per *Fig. 3*

Description (placing)	Cross section of opening		Velocity	
	Designation	Area	Designation	Value
		m^2		m/s
Impeller	A_0	0.0573	c_0	3.21
Pump housing top (pressure tail-piece)	$A_{ny} + A_{bgy}$	0.0591	c_1	3.11
Pressure pipe row of holes	$48A_1$	0.0336	c_2	2.68
Suction pipe row of holes	$44A_1 + 2A_t$	0.0524	c_3	1.79
Bottom of pump housing (suction tailpiece)	$A_{sz} + A_{bgy}$	0.0615	c_4	2.99

2. Velocity in the Hot End Collector

The propeller pump carries 0.814 m^3/s fluid *upwards*. Assuming the system's friction head is in equilibrium with the delivery head of the pump (cf. Clause 4), the cleaning fluid then should pass through the system in its entirety. The velocities shown in *Fig. 3* have been calculated using the measurements in *Table 1* and are listed in *Table 2*. The velocities correspond to the above fluid movement in line with the laws of continuity. On the pressure or discharge end the velocities indicate an intensive 'blowing effect' (c_2) on the collector wall with some turbulence due to the (c_1) velocities acting at a right angle. It can be assumed that at the lower or suction end the relatively slow (c_3) velocities result in considerably 'calmer' flow near the collector wall. Whatever the surface is and that includes the inner lining of the collector, there is a distinct relationship between the turbulence and the degree of surface cleaning. Clearly, the 'blown' surface is cleaned more effectively, than the surface on the 'suction' side.

3. Flow Characteristics of the Impeller

(See *Fig. 5* for definitions).
Volume rate of flow: $Q = 0.184\ \text{m}^3/\text{s}$
Delivery head: $H = 2.01$ m
Pump efficiency – estimate: $\eta = 0.8$
Hydraulic efficiency:

$$\eta_h = \sqrt{\eta} = \sqrt{0.8} = 0.89, \tag{1}$$

therefore the theoretical delivery head is

$$H_e = \frac{H}{\eta_h} = \frac{2.01\ \text{m}}{0.89} = 2.26\ \text{m}. \tag{2}$$

Pump speed, O.D. and hub diameter:

$$N = 980\ \text{rpm}, \qquad D_k = 0.3\ \text{m}, \qquad D = 0.13\ \text{m}. \tag{3}$$

The corresponding peripheral velocity is:

$$u_k = \frac{D_k \pi n}{60\ \text{s/min}} = \frac{0.3\ \text{m}\pi \cdot 980/\text{min}}{60\ \text{s/min}} = 15.39\ \text{m/s}, \tag{4}$$

$$u_a = \frac{D_a \pi n}{60\ \text{s/min}} = \frac{0.13\ \text{m}\pi \cdot 980/\text{min}}{60\ \text{s/min}} = 6.67\ \text{m/s}, \tag{5}$$

the ratio of diameters:

$$\mu = \frac{D_a}{D_k} = \frac{0.13\ \text{m}}{0.3\ \text{m}} = 0.43, \tag{6}$$

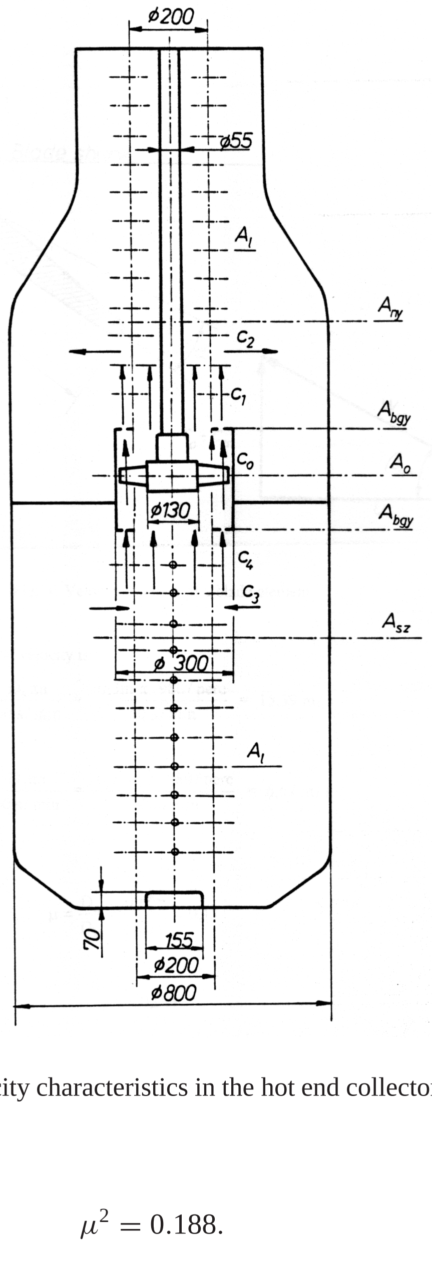

Fig. 3. Velocity characteristics in the hot end collector

$$\mu^2 = 0.188. \tag{7}$$

The meridional velocity assuming zero whirl entry:

$$c_o = c_m \frac{4Q}{(1-\mu^2)D_k^2\pi} = \frac{4 \cdot 0.184\ \mathrm{m}^3/\mathrm{s}}{0.812 \cdot 0.3^2\ \mathrm{m}^2 \cdot \pi} = 3.21\ \mathrm{m/s} \tag{8}$$

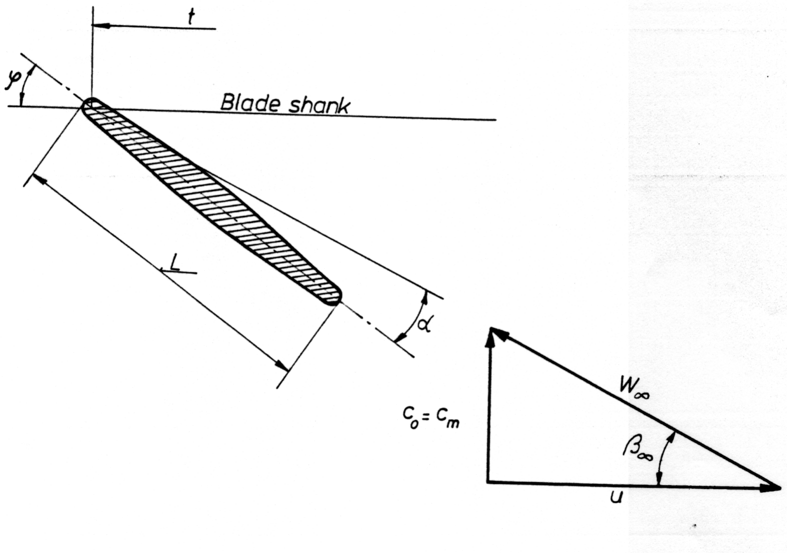

Fig. 4. Velocity triangle of a blade element

change in velocity in the direction of rotation on the circumference and on the hub:

$$\Delta c_{uk} = \frac{gH_e}{u_k} = \frac{9.81\ \text{m/s}^2 \cdot 2.25\ \text{m}}{15.39\ \text{m/s}} = 1.43\ \text{m/s}, \tag{9}$$

$$\Delta c_{ua} = \frac{gH_e}{u_a} = \frac{9.81\ \text{m/s}^2 \cdot 2.25\ \text{m}}{6.67\ \text{m/s}} = 3.31\ \text{m/s}. \tag{10}$$

According to *Fig. 4* the angle of setting is the sum of β_∞: angle of incidence and α: the blade angle:

$$\vartheta = \beta_\infty + \alpha. \tag{11}$$

The calculation of the displacement velocity triangle for the circumference and for the blade root follows (see also *Fig. 5*):

As above

$$\begin{aligned} u_k &= 15.39\ \text{m/s}, & u_a &= 6.67\ \text{m/s}, \\ \Delta c_{uk} &= 1.43\ \text{m/s}, & \Delta c_{ua} &= 3.31\ \text{m/s}. \end{aligned} \tag{12}$$

Thus the velocities of incidence are:

$$w_{ok} = \sqrt{3.21^2\ \text{m}^2/\text{s}^2 + 15.39^2 \text{m}^2/\text{s}^2} = 15.72\ \text{m/s}, \tag{13}$$

$$w_{oa} = \sqrt{3.21^2\ \text{m}^2/\text{s}^2 + 6.67^2 \text{m}^2/\text{s}^2} = 7.40\ \text{m/s}. \tag{14}$$

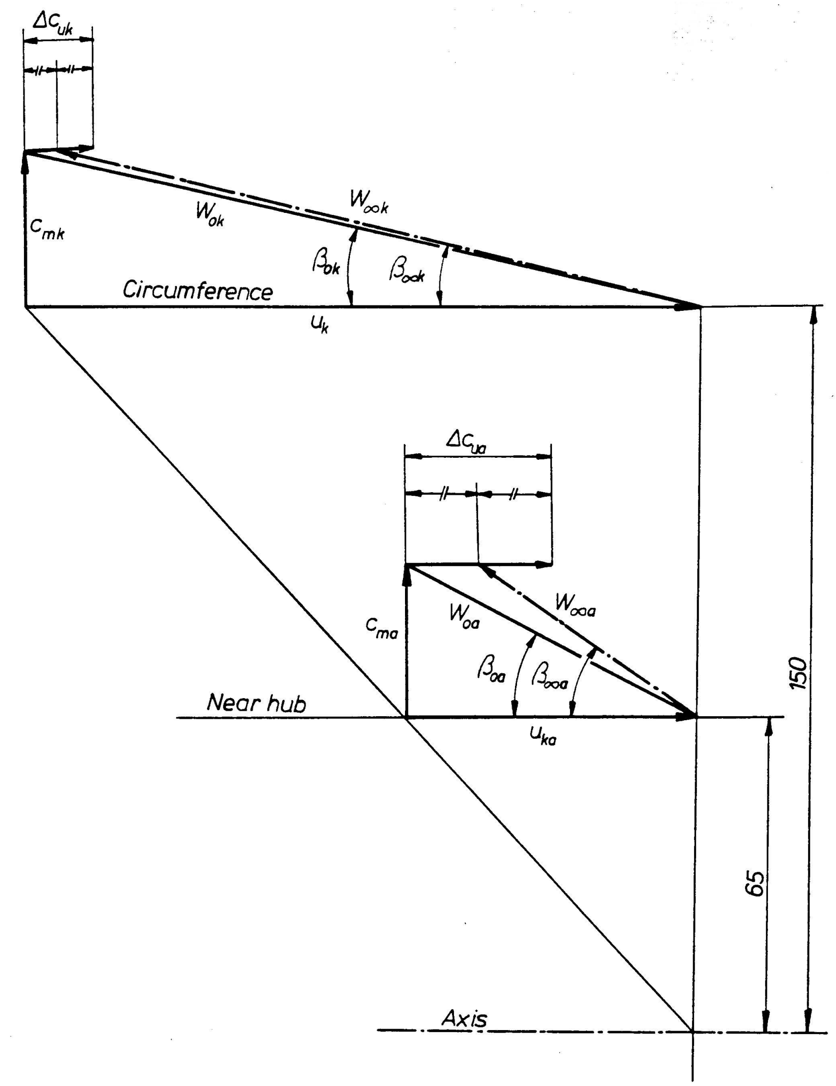

Fig. 5. Displacement velocity triangle of the propeller blade on the circumference and on the blade root

From the entry velocity triangle:

$$\text{tg}\,\beta_{ok} = \frac{c_{mk}}{u_k} = \frac{3.21\ \text{m/s}}{15.39\ \text{m/m}} = 0.209, \qquad (15)$$

$$\text{tg}\,\beta_{oa} = \frac{c_{ma}}{u_a} = \frac{3.21\ \text{m/s}}{6.67\ \text{m/m}} = 0.481, \qquad (16)$$

or

$$\beta_{ok} = 11.8^\circ, \qquad \beta_{oa} = 25.7^\circ, \tag{17}$$

according to *Fig.* 5

$$w_\infty^2 = c_o^2 + (u - \Delta c_u/2)^2, \tag{18}$$

or

$$w_{ok} = \sqrt{3.21^2\ \text{m}^2/\text{s}^2 + 14.67^2 \text{m}^2/\text{s}^2} = 15.02\ \text{m/s}, \tag{19}$$

$$w_{oa} = \sqrt{3.21^2\ \text{m}^2/\text{s}^2 + \ \ 5.1^2\ \ \text{m}^2/\text{s}^2} = \ \ 5.95\ \text{m/s}. \tag{20}$$

In case of zero whirl entry

$$\sin\beta_\infty = \frac{c_o}{w_\infty} \tag{21}$$

and

$$\sin\beta_{\infty k} = \frac{3.21\ \text{m/s}}{15.02\ \text{m/m}} = 0.214, \tag{22}$$

$$\sin\beta_{\infty a} = \frac{3.21\ \text{m/s}}{5.95\ \text{m/m}} = 0.539, \tag{23}$$

$$\beta_{\infty k} = 12.4^\circ, \qquad \beta_{\infty a} = 32.6^\circ. \tag{24}$$

A valid equation for zero whirl entry is

$$\frac{L}{t} c_f = \frac{2gH_e}{w_\infty \cdot u}, \tag{25}$$

where c_f is the lift coefficient of the blade element.

On the circumference and on the hub the LHS products are:

$$\frac{L_k}{t_k} c_{fk} = \frac{2 \cdot 9.81\ \text{m/s}^2 \cdot 2.25\ \text{m}}{15.02\ \text{m/s} \cdot 15.39\ \text{m/s}} = 0.191, \tag{26}$$

$$\frac{L_a}{t_a} c_{fa} = \frac{2 \cdot 9.81\ \text{m/s}^2 \cdot 2.25\ \text{m}}{5.95\ \text{m/s} \cdot \ \ 6.67\ \text{m/s}} = 1.112. \tag{27}$$

Dividing by the three blade shanks on the circumference and on the hub:

$$t_k = \frac{D_k \pi}{3} = \frac{0.3\ \text{m}\pi}{3} = 0.314\ \text{m}, \tag{28}$$

$$t_a = \frac{D_a \pi}{3} = \frac{0.13\ \text{m}\pi}{3} = 0.136\ \text{m}. \tag{29}$$

According to the possible estimates, the length of blade sections should be less than or equal to the result of the division. (The blade shanks do not overlap).

Thus assuming the value of

$$\frac{L_k}{t_k} = \frac{L_a}{t_a} = 1 \tag{30}$$

the lift coefficients are

$$c_{fk} = 0.191 \quad \text{and} \quad c_{fa} = 1.112. \tag{31}$$

These values correspond to lift coefficients of usual asymmetric profiles positioned at certain angles and dimensioned in line with accepted principles of fluid transfer design.

4. Pump Type and Delivery Head

Nominal data of the pump:

Volume rate of flow: $Q = 0.814\ \text{m}^3/\text{s}$

Capacity increase per unit mass: $Y_n = 19.7$ J/kg

Nominal speed $n = 980$ rpm

Drive motor: VP 160 106

3×380 V, nominal power output, $P_l = 11$ kW

Thus the delivery head of pump is

$$H = \frac{Y_n}{g} = \frac{19.7\ \text{m}^2/\text{s}^2}{9.81\ \text{m/s}^2} = 2.01\ \text{m}. \tag{32}$$

Characteristic number of revolutions

$$n_q = nQ^{1/2} \cdot H^{3/4} = \frac{980 \cdot \sqrt{0.184}}{\sqrt[4]{2.01^3}} = 250. \tag{33}$$

This value is well within the usual range of propeller pumps, the estimated efficiency being

$$\eta = 80\%. \tag{34}$$

Thus, the power output can be expressed as

$$P_h = \rho g Q H = 1000\ \text{kg/m}^3 \cdot 9.81\ \text{m/s}^2 \cdot 0.184\ \text{m}^3/s \cdot 2.01\ \text{m}$$

$$= 3630\ \text{W} = 3.6\ \text{kW}. \tag{35}$$

The power input is then

$$P = \frac{P_h}{\eta} = \frac{3.6 \text{ kW}}{0.8} = 4.5 \text{ kW}. \tag{36}$$

Being a direct drive this can be taken as the power output of the motor with a load factor

$$x = \frac{P}{P_t} = \frac{4.5 \text{ kW}}{11 \text{ kW}} = 0.41 = 41\%. \tag{37}$$

It can be concluded that the motor has adequate spare capacity to drive a higher throughput pump, should the need arise.

The delivery head of the device accommodates the load capacity per unit weight (H_B) as well as the losses (h').

$$H = H_B + h'. \tag{38}$$

As in this case

$$H_B = 0, \qquad \text{and therefore} \qquad H = h', \tag{39}$$

thus all of the head is fully available to overcome the resistance of the system. The pressure difference between the two ends of the impeller is:

$$\Delta p = \rho g H = 1000 \text{ kg/m}^3 \cdot 9.81 \text{ m/s}^2 \cdot 2.01 \text{ m} = 19718 \text{ Pa} = 19.7 \text{ kPa}. \tag{40}$$

Considering the propeller ring, this pressure difference results in an additional load

$$F = \Delta p A_o = 19718 \text{ kg/s}^2\text{m} \cdot 0.0573 \text{ m}^2 = 1130 \text{ N}. \tag{41}$$

The base load consists of the weight of the propeller and shaft system plus the Archimedean upward thrust. This latter force is of negligible magnitude. Force F compels the shaft to a pulling action with the aid of the threaded propeller cap. In a suitably modified pump delivering the fluid downwards, the additional axial load, F, would act *upwards*. The inlet and outlet of the pump are separated by removable segments. These segments consist of 800 mm O.D. and 323 mm I.D. rings. Assuming again a pressure difference of Δp acting uniformly over the surface of this ring, the downward force, K, can be expressed as:

$$K = \left(\frac{0.8^2\pi}{4} - \frac{0.323^2\pi}{4}\right) \text{m}^2 \cdot 19718 \text{ kg/s}^2 \cdot \text{m} = 8290 \text{ N}. \tag{42}$$

If the suitably modified pump were to deliver liquid downwards, then force K would act on the ring surfaces in the *upward* direction.

Before modifying the pump as proposed in Clause 5, it will be necessary to evaluate the effect of changing the direction of both the F and K forces.

5. Conclusion

Decontamination should be carried out half of the time(s) under the usual operating conditions, and another half of the time(s) under the reversed impeller and drive direction conditions.

The solution to the task is a new impeller. It should be possible to design and install an impeller featuring symmetrical blades that can be operated in two directions. The cross section of the blade is in accordance with the conditions imposed by the project. The angle of alignment with a constant slope was selected, resulting in pressure heads corresponding to the calculated heads of the existing impeller sections.

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