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TABLE 5 VALUES FOR THE SELECTION OF ROLLER CHAINS

Single-width chains					Triple-width chains				
X	Y	Z	Std. chain no.	Pitch and width	X	Y	Z	Std. chain no.	Pitch and width
2.46	0.185	2662	35N	$3/8 \times 3/16$	7.41	0.555	2662	T35N	$3/8 \times 3/16$
4.00	0.314	2500	41	$1/2 \times 1/4$	12.05	0.942	2500	T41	$1/2 \times 1/4$
4.96	0.427	2310	40	$1/2 \times 5/16$	14.90	1.281	2310	T40	$1/2 \times 5/16$
7.69	0.850	1830	50	$5/8 \times 3/8$	23.10	2.550	1830	T50	$5/8 \times 3/8$
11.90	1.534	1500	60	$3/4 \times 1/2$	35.60	4.700	1500	T60	$3/4 \times 1/2$
20.20	3.470	940	80	$1 \times 5/8$	60.5	10.41	940	T80	$1 \times 5/8$
30.50	6.320	645	100	$1\,1/4 \times 3/4$	91.5	18.96	645	T100	$1\,1/4 \times 3/4$
47.5	11.43	520	120	$1\,1/2 \times 1$	142.0	34.29	520	T120	$1\,1/2 \times 1$
57.8	16.00	370	140	$1\,3/4 \times 1$	172.0	48.00	370	T140	$1\,3/4 \times 1$
80.6	24.95	325	160	$2 \times 1\,1/4$	241.8	74.85	325	T160	$2 \times 1\,1/4$
122.0	51.26	240	200	$2\,1/2 \times 1\,1/2$	366.0	153.78	240	T200	$2\,1/2 \times 1\,1/2$
Double-width chains					Quadruple-width chains				
4.95	0.370	2662	D35N	$3/8 \times 3/16$	9.90	0.740	2662	Q35N	$3/8 \times 3/16$
8.06	0.628	2500	D41	$1/2 \times 1/4$	16.10	1.256	2500	Q41	$1/2 \times 1/4$
9.93	0.854	2310	D40	$1/2 \times 5/16$	19.84	1.708	2310	Q40	$1/2 \times 5/16$
15.40	1.700	1830	D50	$5/8 \times 3/8$	30.80	3.400	1830	Q50	$5/8 \times 3/8$
23.70	3.065	1500	D60	$3/4 \times 1/2$	47.4	6.130	1500	Q60	$3/4 \times 1/2$
40.30	6.940	940	D80	$1 \times 5/8$	80.6	13.88	940	Q80	$1 \times 5/8$
61.0	12.64	645	D100	$1\,1/4 \times 3/4$	121.8	25.28	645	Q100	$1\,1/4 \times 3/4$
95.0	22.86	520	D120	$1\,1/2 \times 1$	188.0	45.72	520	Q120	$1\,1/2 \times 1$
115.7	32.00	370	D140	$1\,3/4 \times 1$	231.5	64.00	370	Q140	$1\,3/4 \times 1$
161.0	49.90	325	D160	$2 \times 1\,1/4$	322.4	99.80	325	Q160	$2 \times 1\,1/4$
244.0	102.52	240	D200	$2\,1/2 \times 1\,1/2$	488.0	205.04	240	Q200	$2\,1/2 \times 1\,1/2$

Substituting $2C + [(N_1 + N_2)/2]$ for L in formula [28*a*], this substitution being sufficiently accurate, we obtain

$$H = 0.023PBN_1N_2[2C/(N_1 + N_2) + 1/2] \quad \ldots\ldots [28b]$$

$$H = 0.0127PBdN_1n_1 \quad \ldots\ldots [29b]$$

The horsepower rating can now be said to be the lesser of the two values computed from formulas [28*b*] and [29*b*]. The effect of a change in center distance is shown in the following example.

Let P = 1 in., N_1 = 15, N_2 = 21, C = 21, and the speed = 547 rpm. Then the rated horsepower is 10.5.

If the center distance is changed to 42 pitches, the rated horsepower is one and two-thirds times as great, or 17.5 hp.

Now from formulas [28*b*] and [29*b*]

$$23PB = \frac{1000H}{N_1N_2[2C/(N_1 + N_2) + 1/2]} \quad \ldots\ldots [28c]$$

$$12.7PBd = 1000H/N_1n_1 \quad \ldots\ldots [29c]$$

The first member of formulas [28*c*] and [29*c*] has a fixed value for each standard *chain*, and the second member has a particular value for each chain *drive*.

In this closure the author gives a table, numbered Table 5 as a continuation of the table numbers given in the paper. In Table 5, column X shows values of $23PB$ for the several standard chains listed in Table 1 of the paper. The column headed Y shows values of $12.7PBd$ for the same chains, and the column headed Z gives the max rpm for a 17-tooth sprocket as listed in Table 1 of the paper.

If we compute

$$x = \frac{1000H}{N_1N_2[2C/(N_1 + N_2) + 1/2]}$$

and

$$y = 1000H/N_1n_1$$

for a proposed drive, and z is taken as the revolutions per minute of the smaller sprocket, it is then only necessary to refer to the table for a set of values of X, Y, and Z, each of which is greater than x, y, and z. The standard chain number is then found in column 4 of Table 5. In most cases there will be several possible selections, and the one highest in the list will usually be the most economical. A quick reference to the table of maximum revolutions per minute will give an adequate check on the selection.

As an example, let it be required to select a chain to transmit 15 hp over sprockets having 15 and 25 teeth, the smaller wheel turning at 900 rpm, and the center distance to be about 30 pitches. Then

$$x = \frac{1000 \times 15}{15 \times 25\{[(2 \times 30)/(15 + 25)] + 0.5\}} = 20$$

$$y = 1000 \times 15/15 \times 900 = 1.111, \quad \text{and } z = n_1 = 900$$

In Table 5, sixth row from top, the values of X, Y, and Z are 20.20, 3.470, and 940, respectively, all of which are greater than the required values. The standard chain number is 80 and is a 1-in. pitch chain. It will be noticed that double-, triple-, and quadruple-width chains are listed as well as the single-width chain. If a selection cannot be made from the list of single chains, reference is made to the double-width chains, then to triple-width or the quadruple-width chains. In this case chain number *D*60 could be used but it would not be as economical as the single-width chain No. 80.

Referring to J. N. Arnold's discussion, the difference in the results obtained from his chart and those obtained by the method just described is due to the fact that the author is here taking account of the effect of a change in center distances, whereas Mr. Arnold's chart was designed to give results corresponding to formulas [28] and [29] and to Table 3. Doubtless Mr. Arnold's chart could easily be modified to take care of varying center distances if desired.

Effect of an Excessive Number of Teeth. One more comment seems to be important. An inspection of Table 4 of the paper shows that in the case of drives having over 60 teeth in the larger sprocket the horsepower ratings seem to run higher than they should. The reason for this is that the present research has dealt only with the rate of chain elongation and has not taken into account the fact that, whereas chains operating over sprockets with less than 60 teeth can often stretch 3 per cent before being discarded, this is not true where the sprockets have considerably more than 60 teeth.

In the case of 90 teeth the chain will climb very nearly to the top of the teeth when the elongation is only 2 per cent; and this means that the allowable life of the chain on a 90-tooth sprocket is only two-thirds as great as one running on a 60-tooth sprocket. To take account of this a note may be attached to Table 5 stating that, where N_2 is more than 60, the values of x are to be multiplied by $N_2/60$ before being applied to the table.

The Leakage of Steam Through Labyrinth Seals[1]

R. L. Scorah.[2] The subject of flow through labyrinth seals has received rather limited study, even though it has been known for sometime that the older formulas do not always give reliable results. Mr. Egli's paper will no doubt form a welcome addition to the literature of this field.

It should be pointed out, however, that the static tests referred to in this paper do not simulate the real conditions of operation. In practice, the leakage steam flows in a different sort of channel, one part of which is stationary while another part, the shaft, rotates at high speed. It has not been demonstrated that the leakage is independent of the speed of rotation. Furthermore,

[1] Published as paper FSP-57-5, by Adolf Egli, in the April, 1935, issue of the A.S.M.E. Transactions.

[2] Mechanical Engineering Department, Stanford University, California.

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the argument presented here is confined to the gas phase of the flowing fluid, that is, superheated steam. In some cases, the steam entering the seal will be wet or only slightly superheated. Under these conditions, the effect of supersaturation and water drops further complicate the problem. In view of the wide variety of designs used in practice and the absence of precise information regarding supersaturation and the behavior of entrained water drops, it would seem expedient at this time to investigate experimentally the leakage through various labyrinth constructions using a rotating-shaft element. Such experiments would not only provide reliable leakage data for the particular constructions tested but would also serve as a helpful check on the rational theoretical treatment of the problem.

H. G. YATES.[3] The writer is interested in knowing if Mr. Egli has studied the effect of inclining the baffle strips at an angle of 45 deg. Glands of this type in conjunction with a spring mounting have been found satisfactory. It is uncertain, however, how far their good sealing properties are due to the inclination and how far to the accurate maintenance of fine clearances by means of the spring mounting. Any gain in sealing properties would probably be proportionately smaller with very fine clearances (of the order of 0.01 in. or less) as the small radius, which necessarily exists on even a "sharp" baffle strip, would assume proportionately greater importance, and the constriction would cease to be a sharp-edged orifice and would tend toward one with well-rounded edges. It is presumably this effect which gives rise to the curves of Fig. 18 of the paper. An important advantage of the inclined baffle strip is that in the event of local heating due to a momentary rub, the strips tend to cockle up away from the shaft rather than to expand toward it.

Another gland which has been employed frequently is of the "straight-through" type but with the baffle strips adjacent to a number of thin closely pitched ribs turned on the shaft or gland sleeve, of different pitch from the strips themselves, so that there is no fear of a straight blow-through if the rotor should be moved in an axial direction. Besides having the property of rapid cooling in the event of a rub, it is believed that the "valleys" between the ribs or pips serve to break up the steam flow and give almost as good a seal as a gland of the well-staggered type. The writer would welcome Mr. Egli's opinion on this.

B. HODKINSON.[4] Mr. Egli refers more than once in his paper to the high velocity through the last throttling and points out that the Fanno line may at this stage be departed from.

A way of allowing for this in the calculation suggests itself. The pressure before the last throttling may be taken as twice the final pressure, and the calculation carried only to this point. In other words, the last throttling could be done away with, and the final pressure doubled. This would raise the calculated figure slightly, except in odd cases, and in the light of some tests at Trafford Park, Manchester, on a grooved valve spindle, would make the answer more reasonable, because we found a greater mass flow than appeared, according to calculation, to be possible.

The odd cases where the calculated discharge would not be increased occur when p_2/p_1 is higher than about 0.2, and then the last throttling probably does not carry sound velocity. Most practical labyrinths operate with p_2/p_1 a good deal less than 0.2. Thus, it would be well to use the above suggested way only when p_2/p_1 equals, say, 0.1 or less.

[3] English Electric Co., Ltd., Rugby, England.

[4] Turbine Experimental Section, Mechanical Engineering Department, Metropolitan Vickers Electrical Co., Ltd., Manchester, England.

AUTHOR'S CLOSURE

The author fully realizes the fact, mentioned by Prof. R. L. Scorah, that the actual flow conditions are not correctly simulated in the static tests referred to in the paper. Some of Friedrich's[5] tests, which were made with a rotating shaft, show the effect of peripheral speed to be rather small. At 200 ft per sec the decrease of leakage due to the speed of the shaft has been found to be 3.4 per cent. For moderate peripheral speeds, therefore, the static tests will not be much in error.

There is a further point with a view to simulating the actual flow conditions, which has been neglected in the static tests. It is the exact shape of the edges of the sealing strips after they have been rubbing in the running turbine. The strip generally becomes burred over and the sealing edge is no longer sharp, as assumed in the static tests, but is in cross-section shaped rather like a mushroom. The characteristic function ψ of such a mushroom-shaped throttling undoubtedly is somewhat different from the function ψ of a sharp-edged orifice shown in Fig. 2 of the paper. It would be principally possible to determine this function by testing a single "mushroom-shaped" strip. Curves φ versus pn/po similar to those of Fig. 7 of the paper could then be constructed with the methods there presented. For most practical purposes, however, the φ curves of Fig. 7 will be sufficiently accurate.

The author has had no experience with the sealing properties of the particular types of baffles described by H. G. Yates, and does not believe that by inclining the labyrinth strips the amount of leakage for a given clearance should descrease measurably. If such glands have proved successful in the turbine, it apparently is due to the maintenance of a fine clearance. The "straight-through" type seal with strips adjacent to a mumber of closely pitched ribs is probably somewhat tighter than the "straight-through" type gland referred to in the paper. It is his opinion, however, that this type of packing will not reach the good sealing properties of a well-staggered labyrinth. The leakage jet issuing from one clearance very easily "bridges" over the narrow "valleys" between the ribs by simply maintaining a series of stationary vortices in each valley.

The method of considering the great pressure drop across the last throttling, as suggested by H. B. Hodkinson may give fairly correct results, although the assumption of the pressure before the last throttling to be twice the final pressure is quite arbitrary. There is, however, no need of treating separately the last throttling when following the methods outlined in the paper. The curves of Fig. 7 and 7*a* automatically take into account the flow characteristics of each throttling in the labyrinth including the last one.

The physical nature of the leakage flow along a grooved valve stem, to which Mr. Hodkinson refers, is quite different from that in a labyrinth packing. Here the loss of kinetic energy due to friction in the narrow space between stem and bushing must be taken into consideration whereas in a labyrinth the friction in the short passage through the throttling gap plays a minor rôle only.

[5] "Untersuchungen ueber das Verhalten der Schaufelspaltdichtungen in Gegenlauf-Dampfturbinen," by H. Friedrich, *Mitt. Forsch. Anst. G. H. H.*, Oct., 1933.

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