Table 2 Effect of oil jet velocity on the scoring resistance

$Q = 9.33 \times 10^{-6} m^3/S$ (560 ml/min), $\eta = 1.0 \times 10^{-4} m^2/S$ (100 cSt)

	Rotational speed of driving gear n_1, rpm	Last non-scoring lever load, N(lbs)		Temperature of scattering oil, K(deg C)(a)		Atmospheric temperature in gear box, K (deg) (b)	
		at $V_0=V_b$	at $V_0=V_b/12$	at $V_0=V_b$	at $V_0=V_b/12$	at $V_0=V_b$	at $V_0=V_b/12$
Engaging lubrication for pitch point	1500	623(140)	534(120)	372.3(99.0)	365.2(92.0)	346.2(73.0)	346.0(72.8)
	3000	311(70)	334(75)	366.2(93.0)	368.2(95.0)	347.7(74.5)	349.4(76.2)
	5500	245(55)	245(55)	369.2(96.0)	372.2(99.0)	350.7(77.5)	351.9(78.7)
Disengaging lubrication for pitch point	5500	200(45)	133(30)	———	———	352.0(78.8)	348.7(75.5)
Disengaging lubrication for driving gear center	5500	289(65)	89(20)	———	———	353.1(79.9)	341.1(68.0)

	n_1, rpm	Last non-scoring lever load, N(lbs)	Temperature of scattering oil, K(deg)	Atmospheric temperature in gear box, K(deg)
Engaging lubrication parallel to tooth trace	1500	489.3(110)	370.7(97.5)	347.4(74.2)
	3000	266.9(60)	363.2(90.0)	347.2(74.0)
	5500	200.2(45)	373.7(100.5)	352.0(78.8)

(a),(b) : Temperature corresponds to the value at last non-scoring load

of working flank of driving gear parallel to the tooth trace from the nozzle at engaging side near the meshing region. The oil jet velocity was chosen to make the oil jet proceed at a distance of the face width. As shown in Table 2, the scoring resistance when oil was supplied parallel to the tooth trace decreases approximately 20 percent and also the temperature of scattering oil and the atmospheric temperature in the gear box are higher in comparison with the case of engaging lubrication directed toward the pitch point. This fact suggests that the arch-shaped oil generated at low oil jet velocity contributes to lubricate the working flanks.

Concerning the case of disengaging lubrication, as shown in Table 2, oil jet velocity greatly affects the scoring resistance, and in the case of oil supply directed toward the driving gear center the effect of oil jet velocity on the scoring resistance is greater than that when oil was supplied toward the pitch point. The reason can be explained by examining the instantaneous behavior of lubricating oil supplied onto the tooth-flank. As known from Fig. 8*(b)*2 and Fig. 8*(c)*2, in the case of disengaging lubrication at oil jet velocity much less than v_b, the oil jetted from the nozzle is scattered in the circumferential direction by the action of tip surface of driving gear and is scarcely supplied onto the working flank, especially in the case of oil supply directed toward the driving gear center. As a result of this, in the case of oil supply directed toward the driving gear center, the scoring resistance in the case of $v_0 = v_{12/12}$ became less than one third of that in the case of $v_0 = v_b$, though the atmospheric temperature in gear box at last nonscoring load was low.

In this study it was recognized that the scoring resistance is greatly affected by oil supply conditions such as oil jet velocity, oil-nozzle position, and its direction. Especially in the case of disengaging lubrication, oil jet velocity is a very important factor on the scoring resistance. Therefore, when the correlations between the experimental results of scoring resistance obtained by various gear machines are discussed, it is important to pay attention to the difference of oil supply conditions. The oil supply state of Ryder test rig under standard condition, that is, disengaging lubrication directed toward the pitch point 4.50×10^{-6} m^3/s (270 ml/min), oil flow through a nozzle of 1.016 mm (0.040 in.), and n_1 = 10000 rpm is not good to be considered for tooth flank lubrication in comparison with the standard conditions of IAE test rig [8]. In order to improve the scoring resistance, oil must be supplied over the whole working flank.

5 Conclusions

In this paper, it has been both analytically and experimentally shown that oil supply condition is an important factor for scoring resistance. The results obtained are summarized as follows:

1 Scoring resistance of gears is greatly affected by oil jet velocity, oil-nozzle position, and its direction.

2 Oil directed toward the center of the driving gear at sufficient velocity ($v_0 \geq v_b$) is better than oil directed toward the meshing region.

3 Oil jet velocity is important when oil is supplied toward the center of driving gear or into the disengaging mesh; however, oil jet velocity is not important when oil is supplied into the engaging mesh.

4 When comparing the scoring results between different gear machines, pay close attention to the difference of oil supply conditions.

5 According to the scoring results obtained by IAE Gear Machine, in the case of $v_0 = v_b$, oil supplied on the disengaging side of the mesh is better than oil supplied on the engaging side of the mesh.

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8 IP 166/68.

DISCUSSION

L. S. Akin[2]

The discusser is very pleased to see that the authors have chosen

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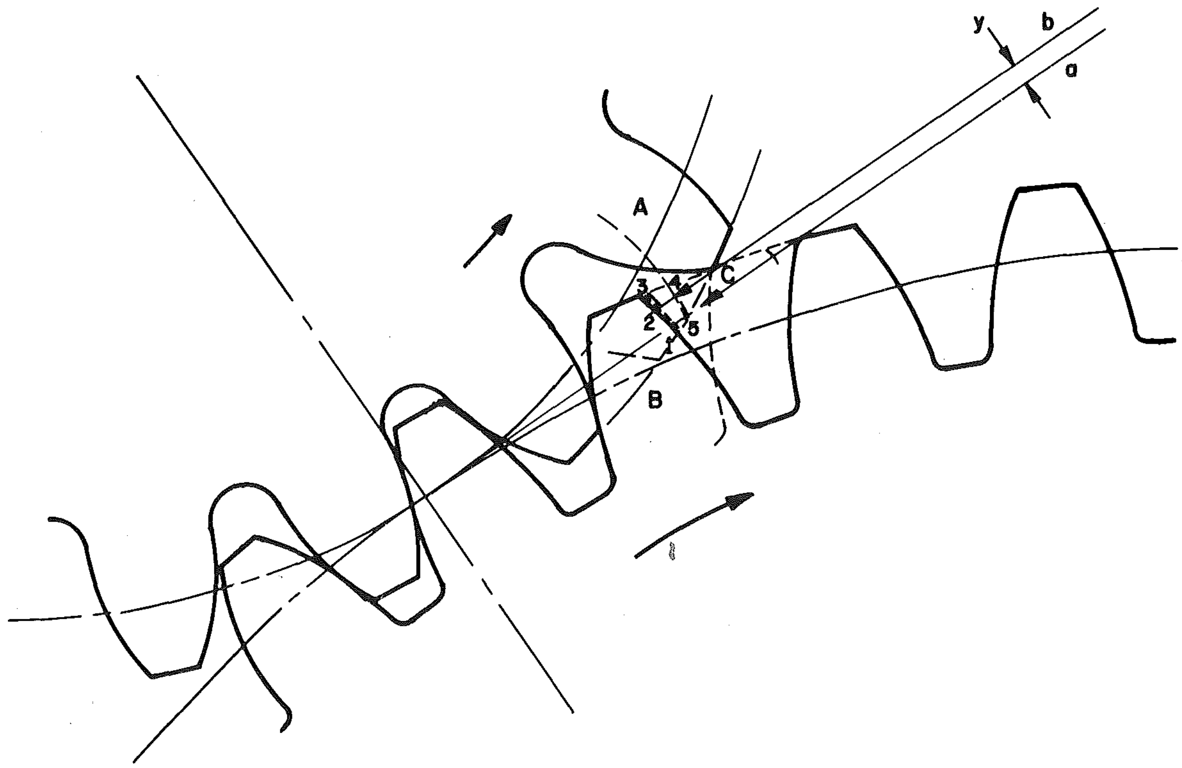

Fig. 10 Tooth interference with oil jet at two different jet positions

to investigate this complex technological area in which too few of us have chosen to do work, perhaps because it is a difficult and costly subject area in which to perform experiments. For this discusser and coauthors working in this same subject area, we welcome this valuable contribution to our bank of knowledge on the subject.

The discusser thinks that the few technical terms that differ from the terms/symbols used in this country can be figured out easily by the reader. For example, when they say that the pictures shown in Figs. 7 and 8 were "one-flash stroboscopic photographs," this discusser presumes they mean that the camera was a single-shot, high-lens speed, *flash camera,* not a synchronized, stroboscopic, high-speed *movie camera,* as was used in our work (reference [9]). Also, the authors use α for the involute pressure angle where we use ϕ and the British use ψ, etc.

The discusser is not going to cover all the items in the paper on which he could comment, in that this would consume too much time. But he will comment on certain specific items in the paper that he thinks need clarification, particularly in relation to large-ratio, very large gears as are used in the marine and industrial or power plant drive systems. Further, he and his coauthors will be producing more papers in this subject area in the near future and will provide additional detail and clarity beyond what time will allow here.

Specific Comments

1 Fig. 3 in the paper shows the *jet stream line* pointed toward the mesh pitch point. In large reduction gears like those used in marine and power plant applications, such a scheme may let the jet miss the *drive pinion "a"* entirely as shown in the discusser's schematic Fig. 10. Jet stream line *"a"* impinges on gear tooth *"b"* only. If the jet stream line is moved over to position *"b"* coinciding with the common intersection of the outside diameters *"c,"* the impingement depth on tooth *b* is reduced from 1–3 to 2–3. But the cooling oil is now shared with the pinion tooth *"a,"* as shown in the dotted position now being cooled from 4 to 5. Generalizing, the jet stream line can be placed at various positions *"y"* from position *"a"* to gain whatever cooling proportionment is desired—for example, forcing all of the coolant to impinge on the pinion instead of the gear. This may be considered desirable in some cases when one wishes the "loaded" or working side of the tooth to be cooled as much as possible. A separate jet directed in a radial or optimum direction can be provided for each gear individually to provide a much better solution, as the authors have asserted.

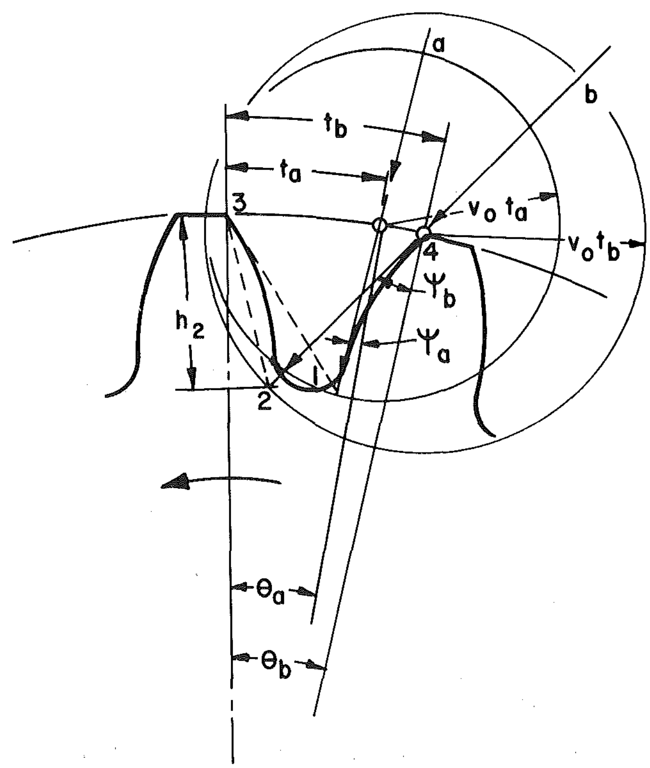

Fig. 11 Kinematics of jet impingement on a tooth

2 In Fig. 4(b) of the paper, the author asserts that with the jet set at any angle ψ between ψ_{max} and ψ_{min}, it is not possible to get the oil to penetrate below Point B at the intersection of the profile tangent $\overline{BO}_0$ and the reverse "line of action" $\overline{SB}$, especially if the jet stream is directed along the line $\overline{SB}$. He further implies (perhaps unintentionally) that it is not possible to impinge directly on the load side of a driven gear. The discusser would like to make the mechanics of this portion of the problem clearer for the reader using his Fig. 11. It is possible to get all the way to the root or to any depth desired (h_v) on either side of the tooth by setting v high enough so that $vt_i \doteq h_v/\cos\theta$ and to the proper angle ψ_i, where h_v, v and θ are defined in reference [10] and the subscript i refers to any specifically selected t and ψ (per the authors' definition).

Two specific examples are shown in the discusser's Fig. 11. In the first example, it is desired to cool the entire loaded face of the driver with the jet directed along stream line "a." Here for given velocities v_o and v_b the jet position angle ψ_a cannot be larger than the angle between the tangent line at the desired depth of impingement (Point 1, in this case) to avoid premature contact with the tooth profile. Also, significantly smaller ψ_i angles would require an increase in the value for v_o in that t_a (time required for gear to rotate through angle θ_a would be smaller and h_a must be maintained at a fixed value to keep the size of the penetration circle a constant. Thus, as the gear rotates relative to the nozzle, the oil jet stream penetrates to a position along the dotted line 1–3, as was shown in the discusser's reference [2] as Fig. 6. This same figure also shows that the loaded (or back) side of the driven gear can also be cooled by jet impingement if the jet stream is directed along the line "b" toward impingement point "2" if the nozzle is set at ψ_b, and v_o is selected so that $v_o t_b = h_b/\cos\psi_b$ = (the authors' r_ℓ to point "2"). In this case, the oil jet stream penetrates to a position along the dotted line 2–3 and terminates as point "2" in the time t_b (while gear rotates through angle θ_b) where the leading force of the trailing tooth chops the jet stream off as shown at "4." Here it will be noted that for the given t_b and v_b the angle ψ_b must be larger than the angle between the tangent to the profile at "4" and the radius vector from the center of the gear through "4". If ψ_b is made smaller, then t_b will have to be reduced and v_o increased in order to continue to intercept the point "2" at the desired impingement depth h_b.

Generalizing, ψ_i should be made as large as necessary, within the above restraints, to keep v_o as small as possible for the impingement depth h_i desired. The fact that v_o varies with the square root of the pressure makes minimizing v_o very important in a high-speed drive. Thus it can be shown that the minimum velocity needed to jet cool the working (loaded) side of the driven gear is:

$$v_{o\,\min} \doteq \frac{h_2 n_i}{30(2/N_i - T_\ell/\pi R_o)\cos\psi_b}\ \text{in./s}$$

where:

h_2 = depth to point "2"
n_i = speed in rpm of specific gear
N_i = number of teeth in specific gear
R_o = outside radius of gear
T_ℓ = width of top land of gear

The authors' results regarding scoring could have been much more impressive at realistic PLV's usually found in large power plant systems.

Two other recent important papers written by this discusser related to the scoring portion of the problem are references [11] and [12].

Additional References

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Author's Closure

The authors wish to thank Mr. Akin for his useful remarks. Before, we photographed instantaneous behavior of lubricating oil with a Hycam movie camera. But the photos were not clear. Therefore, in this paper, photos were taken by one-flash stroboscopic method—namely, the area surrounding the gear box was darkened, the shutter of the camera was left open, a single shot of 1/50,000-s strobolight was flashed, and then the shutter was closed.

With regard to disengaging lubrication of large gears, the authors agree with Mr. Akin's remarks.

In Section 2.3, the authors considered the lubrication of leading flank. Fig. 4(b) shows the state that the tip of the oil jet stream just reached any fixed point B on the leading flank. Point B is the bottom point where an oil supply is desired when the oil supply angle is Ψ. The line BO_0 is not tangent to point B. Equations (15)–(18) can be applicable to any spur gear pair if the oil jet is not obstructed by the mating gear. At $\Psi = \Psi_{min}$, v_{i0} becomes infinite because the tip of

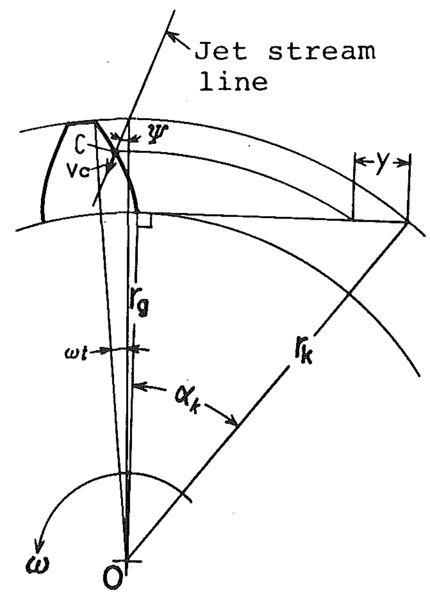

Fig. 12 Velocity v_c of the intersection C of the trailing flank and the jet stream line

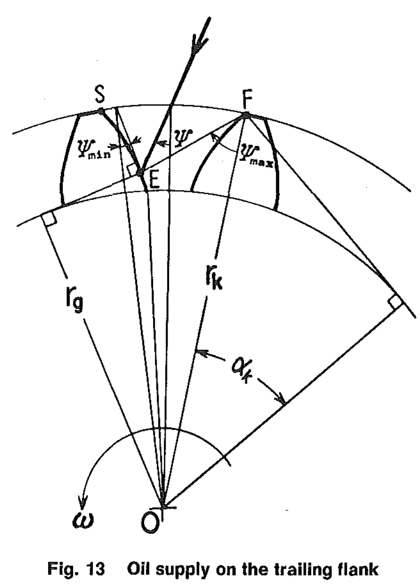

Fig. 13 Oil supply on the trailing flank

the oil jet stream must reach point B at the same time that it passes through point S.

Lastly, the authors explain their view on the lubrication of the trailing flank. In this case, oil jet velocity must be determined by the velocity, v_c, of the intersection C of the trailing flank and by the jet stream line in Fig. 12 when the gear rotates. The velocity, v_c, is given by the following equation:

$$v_c = \omega \cdot \frac{r_g\{(r_g \tan \alpha_k - y)^2 + r_g^2\}}{r_g \cdot r_k \sin \Psi + (r_g \tan \alpha_k - y)\sqrt{(r_g \tan \alpha_k - y)^2 + r_g^2 - (r_k \sin \Psi)^2}} \tag{20}$$

and v_c is monotonically increasing function with respect to y. Therefore, in order to lubricate the trailing flank from tip point S to any fixed point E in Fig. 13, the minimum oil jet velocity $v_{0\min}$ must be equal to the velocity v_c at point E. The oil supply angle Ψ is limited to the range between the angle of $\Psi_{\min}$ and the angle of $\Psi_{\max}$. $\Psi_{\min}$ is the angle which corresponds to the tangent at point E, and $\Psi_{\max}$ is the angle between lines FE and FO. The value of Ψ is defined positive in the clockwise direction. So that the oil jet will not be obstructed by the leading flank, the following inequality must be satisfied:

$$\Psi_{\max} > \alpha_k \tag{21}$$

In the case of $\Psi_{\max} \leqq \alpha_k$, $\Psi_{\max}$ is defined as the angle between lines F′E and F′O where point F′ is the tangent point on the leading flank of the line F′E.

When point E is selected as the point which corresponds to the form circle, $v_{0\min}$ is given by the following equation:

$$v_{0\min} = v_p \cdot \frac{\cos \alpha_c \left[\left\{\sin \alpha_c - \frac{2(1-\chi)}{z \sin \alpha_c}\right\}^2 + \cos^2 \alpha_c\right]}{\cos\alpha_c \left\{1 + \frac{2(1+\chi)}{z}\right\} \sin \Psi + \left\{\sin \alpha_c - \frac{2(1-x)}{z \sin \alpha_c}\right\} \sqrt{\left\{\sin \alpha_c - \frac{2(1-\chi)}{z \sin \alpha_c}\right\}^2 + \cos^2\alpha_c - \left\{1 + \frac{2(1+\chi)^2}{z}\right\} \sin^2 \Psi}} \tag{22}$$

where v_p is the circumferential velocity of reference pitch circle. Table 3 shows the value of $\Psi_{\max}$ and $v_{0\min}/v_p$ at α_c = 20 deg and $\Psi = \Psi_{\max}$. The first column is the value of $\Psi_{\max}$, the second column is the value of equation (22), and the third column is the value of Mr. Akin's equation. The empty column shows the pointed tooth. When undercut occurs, point E is selected as the origin of involute curve. As is shown in the table, $v_{0\min}$ is not affected greatly by the values of z and x, and it is sufficient to take $v_{0\min}$ equal to $1 \cdot 1 v_p$ in any condition. The values of Mr. Akin's equation are small in comparison to the values of the authors' equation. The difference between the two is great when z and x are small.

Table 3 Values of $\Psi_{\max}$ and $v_{0\min}/v_p$ at α_c = 20 deg and $\Psi = \Psi_{\max}$ (v_p = circumferential velocity of reference pitch circle)

Number of teeth		Addendum modification coefficient				
		-1.0	-0.5	0.0	0.5	1.0
14	$\Psi_{\max}$,deg	69.890	57.679	47.375	40.666	
	Authors	0.940	0.975	1.050	1.041	
	Mr.Akin	0.534	0.725	0.815	0.858	
17	$\Psi_{\max}$,deg	69.075	56.692	46.288	40.621	41.178
	Authors	0.945	0.998	1.093	1.049	0.976
	Mr.Akin	0.624	0.779	0.862	0.891	0.861
20	$\Psi_{\max}$,deg	67.767	55.323	45.190	40.519	41.195
	Authors	0.954	1.023	1.096	1.056	0.987
	Mr.Akin	0.693	0.822	0.899	0.916	0.885
30	$\Psi_{\max}$,deg	61.868	50.147	43.020	40.131	40.906
	Authors	1.001	1.080	1.098	1.069	1.014
	Mr.Akin	0.835	0.924	0.967	0.968	0.936
50	$\Psi_{\max}$,deg	52.146	45.259	41.172	39.582	40.265
	Authors	1.053	1.091	1.101	1.082	1.043
	Mr.Akin	0.959	1.003	1.022	1.015	0.989
100	$\Psi_{\max}$,deg	44.963	41.672	39.704	38.979	39.435
	Authors	1.080	1.099	1.103	1.093	1.071
	Mr.Akin	1.037	1.057	1.064	1.057	1.039
200	$\Psi_{\max}$,deg	41.510	39.902	38.938	38.597	38.862
	Authors	1.093	1.103	1.105	1.099	1.087
	Mr.Akin	1.072	1.082	1.085	1.080	1.069
Rack	$\Psi_{\max}$,deg	38.146				
	Authors	1.106				
	Mr.Akin	1.106				