Development of the Focke Helicopter

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The paper gives a description of the methods used in the development of the Focke helicopter. After a short review of the analytical and experimental information regarding the helicopter in existence prior to 1932, the author states the fundamental requirements to be met and indicates the procedure adopted to satisfy them. The analytical discussion is based on the Glauert-Lock theory of the autogiro rotor, and the author shows in outline the various steps in extending the theory to cover the helicopter rotor.

Finally, the construction and testing of the first machine are described, together with its successful flights.

HIGH SPEED, the great gift which the airplane has given man, is also a curse, from which he has not yet been delivered. The inability to stand still in the air, to rise vertically, and to sink vertically, besides making necessary the use of large areas for airports, produces many dangers; dangers in taking off and on landing; dangers of poor visibility, of flight over unsuitable terrain, in mountainous regions, and over extended forests or marshes.

We all know the means which have been employed with great success in the airplane to secure minimum speed, and from the very nature of these methods it is impossible to reduce the speed to zero. This is because the necessary down-wash of air, required to produce lift in the airplane, is produced by the forward speed, and furthermore it is impossible to deflect the air through as much as 90 deg, the ideal but unattainable optimum. The third factor, the mass of air affected, can only be increased by increasing the area of the lifting surfaces, and the extent of such increase in area is governed by structural limitations.

Nevertheless it is easy to see why the airplane was the first successful type of flying machine. The reason is found in its simple construction. In the early days when all was unexplored territory in this uncommon and difficult problem it was a tempting idea to have no moving parts on the flying machine itself other than the control surfaces, and with the airplane, the moving mechanism, outside of the engine, was simple, as the propeller could be mounted directly on the motor shaft.

With the first visible successes in the years 1907 to 1909, there arose a situation, which we can now see to have been extremely regrettable, a situation that created limitations where there should have been no limitations. As soon as one type of construction was successful, that type of machine, the airplane, was demanded exclusively, and attempts at flight along other lines were forgotten. We had to wait from twenty to thirty years before we could see that the practical achievement of one solution was not proof that another solution was impossible. The obstinate adherence to the methods with which the first flights had been made, is in itself understandable, as is also the oversight of the fundamental defects of this first solution.

We know exactly what limitations the airplane has today. We allow for these limitations and adjust ourselves to them. We improve the airplane continually, within the limits of its possibilities. That is right and must be so but, in doing this, we must not forget one thing for the future: We can only achieve new fields of application by going back to fundamental principles and exploring new paths.

When the problem of making aircraft independent of the forward speed is put before us one requirement becomes paramount. We must give the supporting elements their own movement relative to the air. This requirement must be met in practice with the least possible complication. Flapping and feathering hold some technical attractions but, after sober reflection, we must conclude that one of the simplest and theoretically best-known machine elements, the airscrew, is most appropriate for this purpose.

As we all know, the idea of the power-driven airscrew on a vertical axis, is quite old, going back to Leonardo da Vinci who drew up the first direct-lift machine. The interesting subsequent history of the lifting airscrew must be omitted here because of lack of space, but we know that it is only recently that it has progressed beyond the laboratory stage.

Two impulses in quite recent times have greatly increased interest in lifting airscrews. One came from necessity. For many military, geographical-exploration, photogrammetric, and radio purposes a craft is required that can take off and land with little or no run, can hover in the air, and land or climb vertically or at any desired angle. Such a craft also opens up the possibility of extensive private flying because it can be turned and maneuvered at will and does not require a large landing field. With the helicopter, roof and garden landings are no longer Utopian.

The other impulse came from the technical side. De la Cierva has shown with his autogiro, that a large rotating airscrew is a reliable element of support, and he has so stated many times. However, his machine is not a direct-lift machine and, apart from the jump take-off, cannot rise or sink vertically or hover in the air, as the airscrew is not driven by the motor but "autorotates" freely in the air stream. The forward speed is produced in the usual manner by the motor and the propeller. The connection between the lifting airscrew and de la Cierva is only a loose one, and de la Cierva himself refused its possibilities.

Without doubt the autogiro represents an interesting transition between the airplane and the helicopter. It has no better take-off than the airplane, but it can land at a speed of around 40 km (25 miles) per hr, and has a further valuable property in that the rotating blades have their own velocity relative to the air and hence cannot be stalled, with resulting loss of control. The common feature of rotating blades has led to the general appelation for both types of rotating-wing aircraft.

The very existence of the autogiro rotor has provided theoretical information without which the helicopter would have been a long time developing. The British Air Ministry, when examination and tests of the de la Cierva autogiro were under way, commissioned aerodynamic experts like Glauert and Lock to undertake the mathematical investigation of the peculiar phenomenon of autorotation, that is, the free rotation of a large airscrew placed at small angles of attack relative to the air stream.

We shall see later that an extension of the Glauert-Lock theory can be applied to the forward flight of a power-driven helicopter rotor, down to quite small velocities. We must also add that for

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Translated by Alexander Klemin. Mem. A.S.M.E.

Presented by W. Laurence LePage.

Contributed by the Aeronautic Division and presented at the Annual Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, New York, N. Y., December 5–9, 1938.

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the helicopter lifting rotor with a purely axial air stream, that is, a rotor without forward velocity, a series of theoretical calculations and experiments had been carried out by von Kármán, Flachsbart, and Helmbold. There were also methods for obtaining the best thrust distribution along the radius, and a stability investigation by von Kármán which drew pessimistic conclusions. These constitute roughly the scope of the theoretical development of the autogiro rotor and of the direct-lift rotor up to about 1932.

In practice, by 1932 the following performances for the helicopters had been accomplished by Pescara (France), d'Ascanio (Italy), and Oehmichen (France), which were official or at least officially confirmed: distance, about 1 km; duration, about 10 min; height, about 18 meters. There had been no real sustained flight with helicopters up to that date.

On the other hand the autogiro was already in regular use, although its performance was and is today somewhat inferior to that of the airplane. To develop a true helicopter five years ago, without the attendant limitations of the autogiro, was, therefore, to develop an entirely new type of craft, not to improve an already existing machine, and, while inventive effort is worthy, sober reflection upon the aspects of the problem and a thorough investigation of each phase is a far better approach.

Fundamental Requirements of a Helicopter

The fundamental requirements of a direct-lift aircraft are, in order of their importance, the following:

Ability to Make an Emergency Landing After Failure of the Power Plant. This most important basic requirement, which has become self-evident with the airplane, no helicopter had as yet fulfilled, although theory had indicated that it would be possible to convert the lifting rotor into an autorotative system such as had been demonstrated by Cierva. To accomplish this it is necessary that the pitch angle of the blades become smaller than that used with the power-driven lifting system. This involves a mechanical complication which cannot, however, be avoided because an aircraft without the capacity to land smoothly after failure of the motor or the transmission is unthinkable in aircraft practice.

Controllability and Stability. The aircraft must be controllable, at least with normal skill, in all flight conditions including hovering. It must also have static stability about all three axes, and it is desirable to have dynamic stability about these axes. In this question of stability had been the weakest spot of all previous helicopter experiments. Helicopters up until this time had possessed little or no stability, and only by lightning-quick movements of the controls had it been possible to maintain flight and then only for a short time. For other types of helicopters, which later failed of accomplishment because of other defects, it was said that they would have been stable, but the reason for such claims of stability was never clear. Von Kármán's pessimistic predictions regarding stability led many to think that a really practical helicopter was impossible.

The autogiro has given no particular difficulty in this regard at normal speeds but theory and practice have shown that below a speed of about 40 mph, the autogiro becomes dynamically unstable about some axes at least, and to some extent unstable about all three axes. In practice this instability at low speeds had caused no difficulty, because the slowest speeds were employed only for short periods. With the helicopter, the outstanding characteristic of which is its ability to hover, this question of stability must be considered much more carefully. We must have, if not complete static and dynamic stability, at least normal controllability which must be maintained for hours, and under all weather conditions.

General Reliability of Functioning. In this respect the helicopter has had a great weakness. One could not speak of reliability of functioning when the duration of flight was of the order of a few minutes. The nonmoving structural parts are not subject to any greater stresses than are the wings of a normal airplane. The transmission parts must have at least the reliability of the motor, and these transmission parts have of necessity to be smaller than the structural elements of the wing cells. It should be understood that we are speaking of general reliability, in addition to which the requirements for emergency landing, already mentioned, must be immediately available, and must confer the ability, comparable to that of an airplane, to make an immediate forced landing.

Simplicity of Handling by the Pilot. The technical phases of design are only a part, perhaps the smallest part, of any new problem. The design must be such that the average pilot can handle and fly the machine. It is, therefore, essential to lighten the task of the pilot as much as possible. He must be provided with a control system and other operating means to which he is already accustomed in the airplane. At most only a few more manual operations must be demanded of him than are required in the operation of the airplane. The controls must not only operate in the usual directions, but also their movements must be of the usual magnitudes.

Performance Must Be Acceptable. It is of course self-evident that we cannot, at least at first, realize in the helicopter the high-speed performance of the airplane. We must pay a price for the special ability of rotating-wing aircraft in the slow-speed region. But it would reduce the practical value of a helicopter greatly if it could only hover in the air and we must seek, therefore, a performance for the helicopter which is at least comparable with that of the airplane.

Reasonable Maintenance. Naturally, all personnel handling new apparatus must become accustomed to it, yet it must be possible to maintain the helicopter structure in much the same fashion as the structure of a conventional aircraft, and the power plant should be susceptible of the same maintenance as the motor of a conventional airplane.

This summary of the most important requirements of the ideal helicopter, indicates that the solution will not be found in a remarkable patentable solution, by a so-called Egg of Columbus. There is only one path really open: To investigate many very different questions with equal care, and with equal thoroughness, and to keep them all in mind when undertaking the construction of a helicopter. Many of these questions, such as the many stability problems, lie in the region of theory, while others, such as easy handling and control, are in the region of the practical arts. In between lie the difficult constructional or mechanical problems.

In this work, as in some earlier work with a direct-lift machine, the procedure was as follows: First, many comprehensive experiments in the wind tunnel; then full-scale tests of the structural elements; and finally full-scale construction and flight tests with verification of every step with scale models. I have found that this took relatively the shortest time, and also assured the greatest success.

Let us summarize what has been said before. Prior to 1932 we had the practical autogiro, the rotors of which were not motor-driven, which afforded low minimum speeds; and, hardly beyond the laboratory stage of development, the helicopter. We saw that, for hovering, for prolonged steep-angle or vertical climbing, and for taking off and landing in the smallest possible space, the helicopter solution was still to come and would have to be combined wth autorotation as an emergency landing feature.

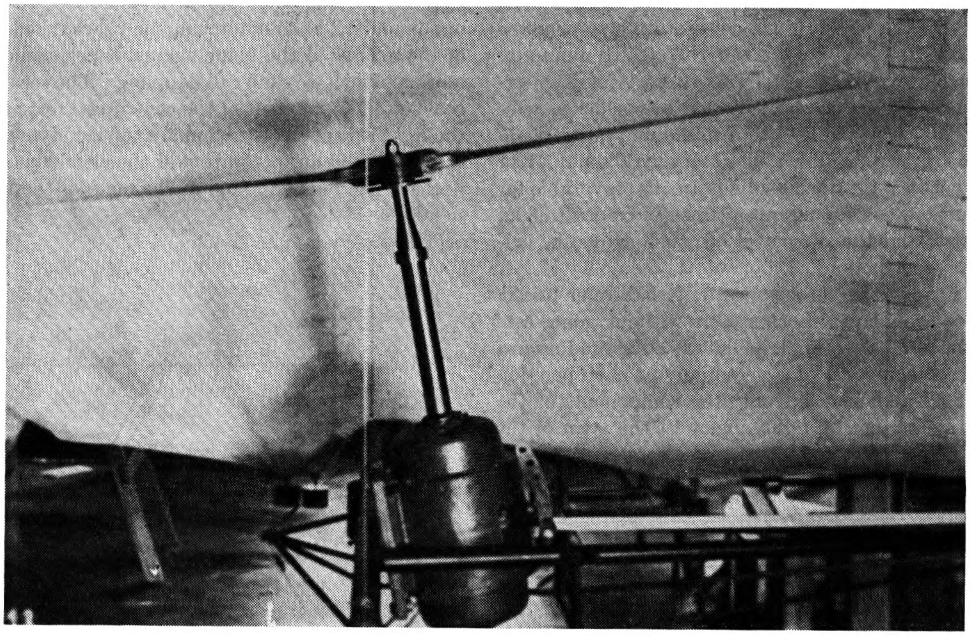

FIG. 1 MODEL OF THREE-BLADED LIFTING ROTOR WITH TEST MOUNTING IN WIND TUNNEL

LIFT PRODUCTION WITH POWER-DRIVEN AND AUTOROTATING AIRSCREWS

The creation of thrust, which is here identical with the production of lift, by an airscrew offers nothing new and I shall assume that the facts relating to the creation of lift or thrust are well known.

In the first place we must consider economy of power. We must seek the best thrust distribution along the radius of the rotor for the least power and the greatest thrust. Some compromise is necessary here, because of our fundamental requirement of emergency landing. The rotor must always autorotate when the power plant fails. If, because of this requirement, some loss in efficiency is incurred, this need not be disturbing.

At the beginning of my own helicopter investigations six years ago, the blades of all successful autogiro rotors, whose autorotation had been demonstrated, were flat (not twisted). If one did not wish to abandon a good thrust distribution along the radius, then the only method was to use strongly tapered blades, of trapezoidal form. These blades were connected to the hub by Cardan mounting, as Colonel Renard had already done in 1904 and as is current practice in most of today's autogiros and helicopters, so as to relieve them of bending moments at the root.

Naturally, these blades were constructed only after much calculation and many measurements in the wind tunnel, both as power-driven and as autorotational systems. Fig. 1 shows a model of the three-bladed screw, driven by a 3-hp electric motor, suspended from the wind-tunnel balance, which measures all forces and moments. Fig. 2 shows some of the test results. The nondimensional power coefficient k_d and also the thrust coefficient k_s are plotted against the pitch angle of the blades ϑ in degrees. According to Bendemann the efficiency of the screw with rotary motion only is given by $(k_s)^{3/2}/2k_d$. The best value of this expression is found to be when ϑ equals 12 deg, and has the value 0.71. That is a good value in spite of the flat (i.e., not twisted) blades. Let us consider the same airscrew, turning freely as an autogiro rotor. The rotor is now started by the motor, the wind tunnel is put into operation, and the rotor is set into autorotation when a certain wind velocity is reached, since a free-wheeling device operates, and disconnects the motor.

The polars can now be measured, just as in the case of the

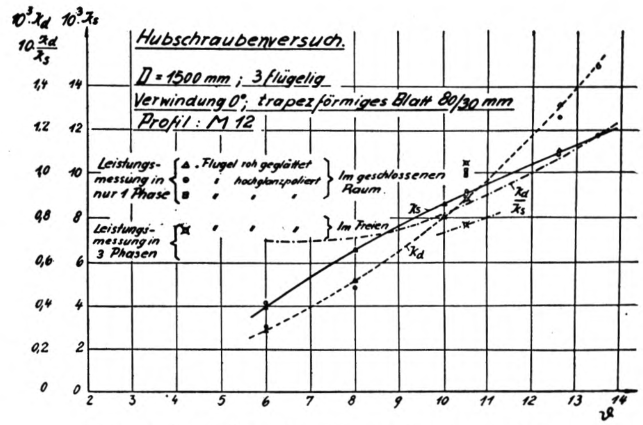

FIG. 2 SOME TEST RESULTS ON MODEL OF FIG. 1

(Power coefficient k_d and thrust coefficient k_s are plotted against the pitch angle ϑ of the blades.

Diam = 1500 mm; three blades, not twisted, tapered from a width of 80 to 30 mm.

Performance measurement in only one phase (of motor); (triangle) blades with rough polish; (circle, square) blades with high polish; in closed space.

Performance measurement in three phases; blades with high polish; unenclosed.)

autogiro. The air speed, the pitch setting of the blades, and the angle of attack of the plane of the rotor, which is termed α_π can be varied. Fig. 3 shows an example of the many polars which have been obtained with a blade pitch of $\vartheta = 4$ deg. The dotted curve in this figure will be referred to later. If the value of ϑ is increased, autorotation is endangered at a smaller value of α_π, that is, in rapid flight. When the value of ϑ is decreased, then the rotor turns faster for the same value of a_π and the polar curve becomes worse. If we keep a pitch setting of $\vartheta = 4$ deg, then we see that for transition from the helicopter rotor, for which $\vartheta = 12$ deg, to the autogiro rotor, the blades must have a pitch decrease of 8 deg, that is to say, the pitch decrease is of appreciable magnitude.

It has been assumed that the choice of the diameter of the rotor has been made beforehand. It is well known that from the point of view of the helicopter rotor, the large slowly revolving rotor is preferable. The diameter is limited only by constructional difficulties. Again the ability to land in an emergency is the deciding factor. The polars of the autorotating rotor go to a maximum value of $C_a(= C_L) = 1$. If the steady minimum speed is placed at 50 km (31 miles) per hr (from American experiments the "unsteady" value of maximum lift coefficient goes to $C_a(= C_L) = 2$ which would lead to a minimum speed of 35 km (22 miles) per hr), we can use a disk loading of about 35 kg per sq m (7.18 lb per sq ft). Such loadings are also favorable for the direct-lift condition, as they permit a thrust of from 7 to 8 kg (15 to 18 lb) per hp, and the diameters are such as can be embodied in practical construction.

For the calculations of the autogiro rotor, I shall refer briefly to the principles of the Glauert-Lock papers without going into the whole development. See Fig. 4. Glauert adds the forward speed v and the rotational speed $R\omega$ vectorially at each position along the radius, and assumes that only the components of the resultant velocity which are at right angles to the span of the

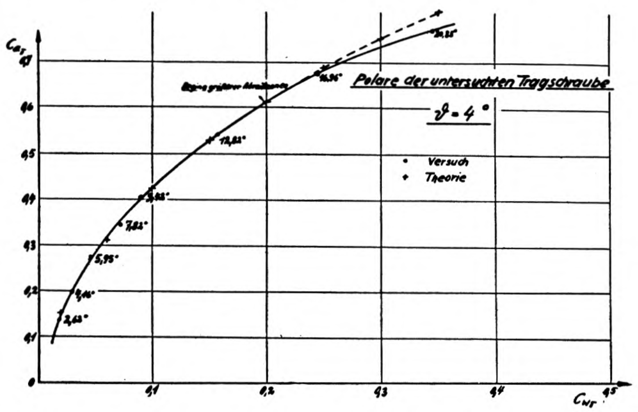

FIG. 3 EXPERIMENTAL AND THEORETICAL POLAR CURVES OF AUTOGIRO ROTOR

(Ordinates, lift coefficient; abscissas, drag coefficient; pitch angle of blades, 4 deg; degree figures on curve refer to angle α_π; dotted curve, theoretical; solid curve, experimental.)

blade need be taken into account for the computation of the air forces. Naturally the local angles of attack change in accordance with the momentary position of the blades during rotation. He designates this local angle of attack by ψ and calculates from the rearmost position of the blades as a base.

In the calculations he employs the air velocity in the form of two parameters, an axial parameter $x = v_d/R\omega$, where v_d is the axial velocity through the plane of the airscrew disk, and also a tangential parameter $\mu = v \cos \alpha_\pi/R\omega$ where $\cos \alpha_\pi$ occurs due to the inclination of the plane of the rotor with reference to the air stream. As the hinged blades are free to take up an equilibrium position, which is the resultant of the weight, the centrifugal force, and the aerodynamic force, and, therefore, bear relationship to the thrust or lift, Glauert is obliged to take the motion of the blades into his calculations. The hinging of the blades allows a forward flight of the rotor, free from the rolling moments which would be present on a rigid rotor, because of the greater lift on the advancing blade. The centrifugal force alone would keep the blades in a plane perpendicular to the axis of rotation as in Fig. 5 (a). Under the influence of the thrust the blades rise upward through an angle β and describe an inverted conical surface, of which the angle β may be called the "coning angle."

Glauert sets up the equation of equilibrium of the masses and aerodynamic forces about the hinge and secures the equation of Fig. 5 (e)

$$I_1\left(\frac{d^2\beta}{dt^2} + \beta\omega^2\right) = \int_0^R rdS_1 \ldots\ldots\ldots\ldots [1]$$

where I_1 is the moment of inertia of a blade, and S_1 is the thrust of one blade. The first term in the bracket indicates the moment of the masses of the blade against a perpendicular up-and-down motion, which we shall call flapping. The second term within the bracket is the moment of the centrifugal force of the blades. On the right side of the equation are the aerodynamic-force moments in the form of an integration of the elements of thrust. Glauert neglects the moments due to the mass of the blades, as they are

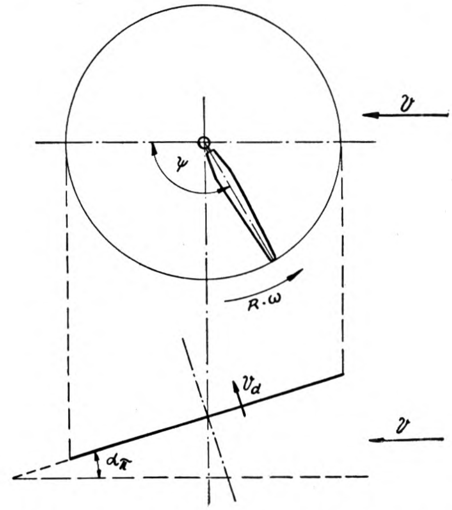

FIG. 4 DIAGRAM OF NOTATION FOR AUTOGIRO ROTOR

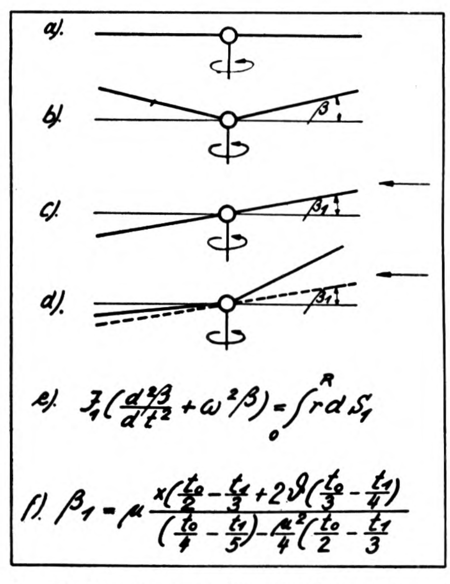

FIG. 5 DIAGRAM FOR GLAUERT-LOCK THEORY

small compared with the moments of the air forces and centrifugal forces. He then develops β and $\int rdS_1$ in a Fourier series and arrives, after some tricks and neglect of terms which I shall not discuss, at the result that the whole cone of the revolving blades is directed backward through an angle β_1 so that β changes periodically with the cycle of revolution, has its maximum value when the blade is in the foremost position, i.e., when $\psi = 180$ deg (Fig. 4), and its smallest value in the rearmost position when $\psi = 0$, when it coincides with the coning angle β. See Fig. 5 (d).

Glauert gives formulas only for rectangular blades, with a constant chord t. The whole Glauert-Lock theory was extended by my co-worker Bansemir for trapezoidal blades, in which extension $t = t_0 - (t_0 - t_1)(r/R)$. Here t_0 is the chord at the root of the blade and t_1 that at the blade tip. The chord decreases proportionately to the radius of the blade. After carrying through the integration of the right side of Equation [1] we obtain, as in Fig. 5 (*f*)

$$\beta_1 = \mu \frac{x\left(\frac{t_0}{2} - \frac{t_1}{3}\right) + 2\vartheta\left(\frac{t_0}{3} - \frac{t_1}{4}\right)}{\left(\frac{t_0}{4} - \frac{t_1}{5}\right) - \frac{\mu^2}{4}\left(\frac{t_0}{2} - \frac{t_1}{3}\right)} \qquad [2]$$

The tipping of the whole blade-system cone signifies, therefore, that the advancing blade rises and the retreating blade falls. Now we secure an important insight into the reason why the rotor with hinged blades experiences no tipping moments about the longitudinal axis of the aircraft. The advancing blade, because of its rising path in space, operates at a smaller angle of attack, and thus the extra lift due to the vectorial addition of the forward speed and the rotational speed is eliminated. The converse is true with the retreating blade. The regulating cause of this phenomenon is found in the centrifugal force which, at the same speed of revolution and with the same weight of all blades, must be the same everywhere.

$$M = z\frac{\rho}{2}C_a' R^4 \omega^2 \Big[\frac{\delta}{C_a'}\left(\frac{t_0}{4} - \frac{t_1}{5}\right) + \frac{1}{2}\frac{\delta}{C_a'}\mu^2\left(\frac{t_0}{2} - \frac{t_1}{3}\right) - \vartheta x\left(\frac{t_0}{3} - \frac{t_1}{4}\right) - x^2\left(\frac{t_0}{2} - \frac{t_1}{3}\right) - \frac{1}{2}\beta_1^2\left(\frac{t_0}{4} - \frac{t_1}{5}\right) - \mu x \beta_1\left(\frac{t_0}{2} - \frac{t_1}{3}\right) - \frac{3}{8}\mu^2\beta_1^2\left(\frac{t_0}{2} - \frac{t_1}{3}\right)\Big]$$

$$M = z\frac{\rho}{2}C_a' R^4 \omega^2 \Big\{\frac{\delta}{C_a'}\left(\frac{t_0}{4} - \frac{t_1}{5}\right) + \frac{1}{2}\frac{\delta}{C_a'}\mu^2\left(\frac{t_0}{2} - \frac{t_1}{3}\right) - \vartheta x\left(\frac{t_0}{3} - \frac{t_1}{4}\right) - x^2\left(\frac{t_0}{2} - \frac{t_1}{3}\right) - \left(\mu \frac{x\left(\frac{t_0}{2} - \frac{t_1}{3}\right) + 2\vartheta\left(\frac{t_0}{3} - \frac{t_1}{4}\right)}{\left(\frac{t_0}{4} - \frac{t_1}{5}\right) - \frac{\mu^2}{4}\left(\frac{t_0}{2} - \frac{t_1}{3}\right)}\right)^2\left[\frac{1}{2}\left(\frac{t_0}{4} - \frac{t_1}{5}\right) + \frac{3}{8}\mu^2\left(\frac{t_0}{2} - \frac{t_1}{3}\right)\right] - \mu^2 x \frac{x\left(\frac{t_0}{2} - \frac{t_1}{3}\right) + 2\vartheta\left(\frac{t_0}{3} - \frac{t_1}{4}\right)}{\left(\frac{t_0}{4} - \frac{t_1}{5}\right) - \frac{\mu^2}{4}\left(\frac{t_0}{2} - \frac{t_1}{3}\right)}\left(\frac{t_0}{2} - \frac{t_1}{3}\right)\Big\}$$

$$x_{Freilauf} = \frac{-\mathfrak{B} + \sqrt{\mathfrak{B}^2 - 4\mathfrak{A}\mathfrak{C}}}{2\mathfrak{A}}$$

$$\mathfrak{A} = BD^2 + B^2D\mu^2 + \frac{3}{16}B^3\mu^4 \qquad B = \frac{t_0}{2} - \frac{t_1}{3}$$

$$\mathfrak{B} = \vartheta CD^2 + \frac{1}{2}\vartheta BCD\mu^2 + \frac{17}{16}\vartheta B^2C\mu^4 \qquad C = \frac{t_0}{3} - \frac{t_1}{4}$$

$$\mathfrak{C} = -\frac{\delta}{C_a}D^3 + 2\vartheta^2C^2D\mu^2 + \left(\frac{3}{2}\vartheta^2BC^2 + \frac{3}{16}\frac{\delta}{C_a}B^2D\right)\mu^4 \qquad D = \frac{t_0}{4} - \frac{t_1}{5}$$

FIG. 6 CHARACTERISTICS OF AUTOGIRO ROTORS

Having obtained β_1 the tipping angle, we can determine the further aerodynamic characteristics of the rotor. See Fig. 6. For the rotational moment which with the autogiro rotor must be zero, Glauert deduced the axial parameter x, the thrust S_1, and the distribution of thrust over the radius of a blade dS_1/dr. See Fig. 7. Finally, he deduced the longitudinal force H, which plays the part of the tangential force of an airfoil, and so lies in a plane perpendicular to the axis of rotation and in the direction of flight.

The rest is simple. Out of the normal force, which is in this case the thrust, and out of the longitudinal force, which is the tangential force, we can derive C_a and C_w, the lift and drag coefficients of the whole rotor. Then the polar diagram is available, of which Fig. 3 is an example. Mention has already been made of the wind-tunnel measurements. The dotted curve of Fig. 3 shows how well the English theory, in spite of its various assumptions and simplifications, agrees with the experimental results.

There is only one thing that the theory does not point out. We cannot predict our trial in terms of our first fundamental requirement, the ability to make a safe emergency landing. The dangerous possibility of cessation of autorotation is not indicated by the theory because it assumes a linear slope for $dC_a/d\alpha$ and autorotation ceases precisely because the greater part of the blade, in the region of stalling, assumes large drag coefficients coupled with small increase in lift. There is a factor of safety in the Reynolds numbers, smaller than occurs with the airplane, at which wind-tunnel experiments are made, if the boundaries of autorotation are determined in this manner. The danger zone lies in quite a narrow region of the pitch settings, With medium blade thicknesses and values of ϑ below 4.5 deg. measured from the angle of no lift, there is no danger of loss of autorotation. In the region $\vartheta = 6$ deg, there is always great danger, as unfortunate accidents have shown, particularly at high speeds and low disk loadings. We note also that the cessation of autorotation behaves in exactly the opposite fashion from the stalling of an airplane.

$$\frac{dS_1}{dr} = \frac{\rho}{2}C_a'\omega^2 t\Big[xrR + \vartheta r^2 + \frac{1}{2}\mu^2\vartheta R^2 + \left(x\mu R^2 - \beta_1 r^2 + 2\vartheta\mu r R + \frac{1}{4}\mu^2\beta_1 R^2\right)\sin\psi + \left(\mu\beta_1 rR - \frac{1}{2}\vartheta\mu^2R^2\right)\cos 2\psi + \frac{1}{4}\mu^2\beta_1R^2\sin 3\psi\Big]$$

$$S = z\frac{\rho}{2}C_a'R^3\omega^2\left[x\left(\frac{t_0}{2} - \frac{t_1}{3}\right) + \vartheta\left(\frac{t_0}{3} - \frac{t_1}{4}\right) + \frac{\mu^2\vartheta}{2}\left(t_0 - \frac{t_1}{2}\right)\right]$$

$$H = z\frac{\rho}{2}C_a'R^3\omega^2\Big[\frac{\delta}{C_a'}\mu\left(\frac{t_0}{2} - \frac{t_1}{3}\right) - \frac{1}{2}\mu\vartheta x\left(t_0 - \frac{t_1}{2}\right) + \frac{3}{2}\beta_1 x\left(\frac{t_0}{2} - \frac{t_1}{3}\right) + \beta_1\vartheta\left(\frac{t_0}{3} - \frac{t_1}{4}\right) + \frac{1}{2}\beta_1^2\mu\left(\frac{t_0}{2} - \frac{t_1}{3}\right)\Big]$$

FIG. 7 CHARACTERISTICS OF AUTOGIRO ROTORS

In our problem of a helicopter that possesses the emergency landing features of an autogiro the hazard of cessation of autorotation plays no great part. If there were any doubt as to the continuation of autorotation all that we would have to do is to reduce the pitch angle ϑ somewhat. Then we shall have completely satisfied our first fundamental requirement.

We have so far considered the production of lift only for the lifting rotor in a stationary position, and as an autogiro rotor. I have thus far purposely neglected the forward flight of helicopter airscrews. Lock has already begun to extend the Glauert considerations to this flight condition. We shall, therefore, pattern the calculations for the helicopter airscrew, in forward flight, on those of the autogiro rotor and then discuss the experiments in this condition. We only need to introduce the moment term about the axis of rotation and we have the helicopter instead of the autogiro rotor. All other considerations remain valid, and the calculations retain the same procedure. Instead of the customary lift coefficients we utilize thrust coefficients as, in the helicopter condition of hovering flight, the coefficient C_a becomes infinity. It is the relationship between the forward flight of the helicopter and its hovering flight, that is particularly interesting. According to the procedure adopted by Flachsbart, we shall, instead of C_a, take the vertical component of the thrust K_{sv}. We shall refer also to the circumferential velocity instead of the forward velocity, and substitute the horizontal component of the thrust K_{sh} for C_w.

In the helicopter condition the value x changes its sign. The flow through the rotor is downward and this remains true in forward flight. The angle of the rotor to the flight path α_π (or in the wind tunnel to the direction of the air stream) has no significance in hovering flight, but in translational flight it can have any value from zero to plus or minus 180 deg, since backward flight is also possible. It is noteworthy that, with the slowest vertical climb, it has a value of —90 deg and, with the slowest vertical descent, a value of +90 deg. In the forward flight of the helicopter, in which there is no other means of propulsion, such as a tractor propeller, the angle of attack α_π must always be negative, as the axis of rotation must be tilted forward, in order to achieve a sufficient forward component to overcome parasite drag.

The results of the calculations for forward flight, will be compared later with the results of experimental work. The measurements of forward-flight conditions were made with the same apparatus as that employed in testing the autogiro rotor. The motor was coupled, and the performance calculations were made with the aid of electrical readings of the power required. The tests also included tachometer measurements of the rotational speed. Great care was required, as there were many chances of

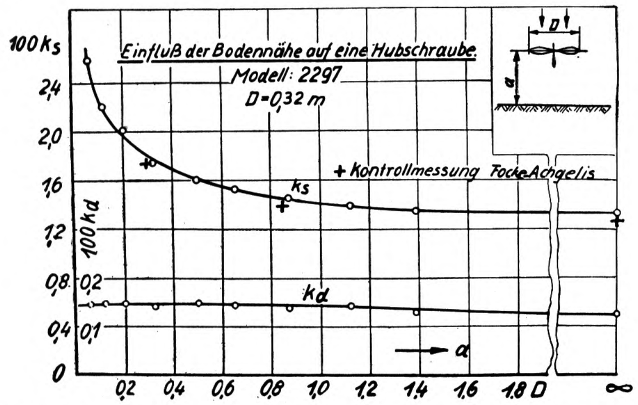

FIG. 8 INFLUENCE OF GROUND EFFECT ON A LIFTING AIRSCREW
(k_s = thrust coefficient; k_d = power coefficient.)

error, which I shall not discuss in detail. I will mention, however, a single factor, the presence of the ground influence. When the ground is level over a sufficiently large area, the thrust increases, as does also the torque. This effect becomes apparent within a height above the ground equal to the rotor diameter. Thus, with a certain throttle setting, the helicopter will rise, but will not rise over a few meters without increase in the throttle setting. A helicopter with too little excess power does not rise over this "swimming level." On the other hand this phenomenon results in a welcome help on landing. In the laboratory it is unpleasant because one can never work far from rigid objects. Up to the present, few experiments have been made on the influence of ground proximity, of which the best ones are those made by Flachsbart in 1928, which our own measurements confirmed rather well. The upper curve of Fig. 8 shows this increase of thrust and the lower curve the increase in torque plotted against the distance from the ground expressed in diameters of the rotor. The three crosses indicate our own control measurements.

CONTROL AND STABILITY

No control is conceivable, unless we first of all establish an equilibrium of moments. There exists with the helicopter the well-known difficulty that in part has influenced the whole question of the direct-lift machine. The slowly rotating airscrews exercise on the aircraft a turning moment of the order of magnitude of thousands of kilogram-meters. The manner in which this torque is removed influences the whole constructional aspect of the helicopter. Many forms have been suggested as Fig. 9 indicates: (*a*) Two screws, one over the other, rotating in opposite directions (Bréguet, d'Ascanio, Pescara, Asboth); (*b*) two screws, one behind the other (Cornu), and also four placed at the corners of a square (de Bothezat, Oehmichen); (*c*) two screws rotating in opposite directions, and placed side by side (Berliner, Focke); (*d*) peculiarly enough, two screws rotating in the same direction, with axes of rotation so disposed relative to one another, that the resulting side moments have been removed (Florine, Belgian government); (*e*) a single large screw on the blades of which small screws are placed (Isacco, Curtiss-Bleeker); (*f*) on which the blades themselves perform oscillations so that a moment does not arise at all; (*g*) a single airscrew, and at the end of the fuselage one or more propellers whose thrust counteracts the torque (Baumhauer, Holländisch Ryksstudiedienst); (*h*) a propeller blowing air backward against deflector surfaces (Hirtenberger Patronenfabrik, Austria); (*i*) surfaces in the slip stream of the helicopter rotor (Hafner and Nagler, Austria); and finally (*k*) a jet-reaction orifice with smaller screws and compressed air (Dornier patent, Papin and Rouilly, France).

From all these proposals we can separate out those which add appreciable drag or loss of efficiency. These are: (*e*) In which the efficiency of the small propeller reduces the efficiency of the helicopter as a whole, by about 30 per cent; (*g*) and (*h*) in which, for the production of the antitorque force, there is needed a continuous expenditure of power, which is of the order of

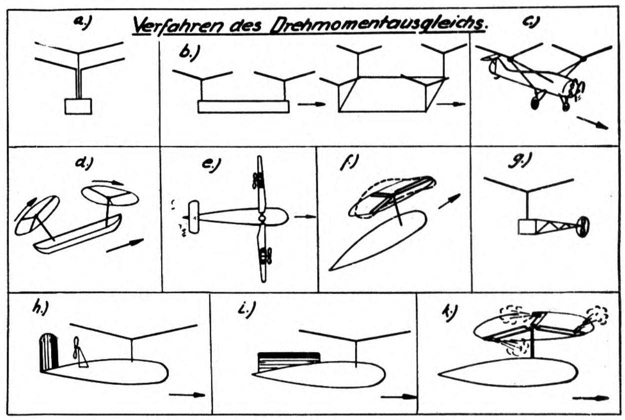

FIG. 9 PROPOSED METHODS OF BALANCING THE TORQUE IN A HELICOPTER

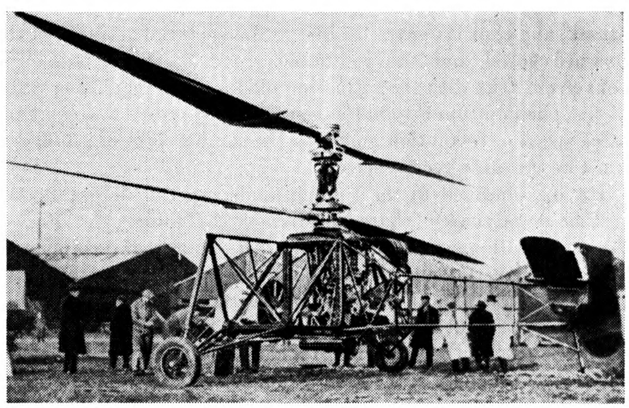

FIG. 10 BRÉGUET-DORAND HELICOPTER

from 20 to 30 per cent; (*f*) and (*i*) reaction and flapping drive—both are perhaps destined to play a part later on; (*d*) the inventors of which wished to retain the gyroscopic effects which are lost with rotors turning in opposite directions, offers no advantages as compared with (*b*) and (*c*). When we eliminate the use of more than two airscrews because of mechanical complication, then there remain only three cases; two screws, placed over each other, behind each other, and side by side, rotating in opposite directions. The first of these three arrangements has been most frequently employed hitherto. The helicopter which was most successful up to the last year, that of Bréguet-

Dorand in France has this arrangement as shown in Fig. 10. The disadvantages of this type are:

(a) Excessive vibration is set up by the passage of the blades past each other.

(b) The efficiency of two rotors in line is much less than with separate rotors.

(c) The slip stream blows over the whole structure so that the effective thrust is further reduced, and the longitudinal balance is affected.

(d) The saving in weight of the arrangement is balanced out by having a smaller disk area for emergency landing.

The only arrangement which entails no prejudicial interferences is the form in which rotors are disposed side by side. Vibrations of the blades disappear, both rotors offer their full disk area in emergency vertical descent, and they mutually provide high lateral stability. The efficiency is as high as that of the single rotor, and in the down-wash of each there lie only the most necessary elements of the structure. Also the difference in space requirements is not much, since what is saved in span by using two rotors placed over one another, is balanced by the greater length of the machine and above all by the greater height.

Now we can consider the stabilizing and control problems. Many helicopters have been controlled and stabilized by means of control surfaces after the manner of the airplane but, in the hovering condition, this is not possible. We can indeed think of surfaces or control surfaces placed in the slip stream of the lifting airscrew rotor, or in the slip stream of the ordinary propeller. We have ourselves tried both with little success. More suited to the nature of the helicopter is a control and stabilizing system through the action of the blades themselves. We must go back here to theoretical calculation and consider next longitudinal stability and control.

(a) *Longitudinal Stability and Control*

The tipping of the blades β increases, as Equation [2] showed, with the tangential parameter μ and, therefore, on the whole, with the speed. As the longitudinal blade pull is practically the only force which can be put into the hinge at the hub and as this force is the same in all blades, it is clear from considerations of symmetry that the parallelogram of forces is a rhombus the diagonal of which is the resultant direction of force. See Fig. 11. It lies at an angle β_1 behind the actual axis of rotation. In other words, there is a virtual rotational axis which slopes downward and backward from the blade hub at an angle β_1 with the actual axis. This is only approximately correct, as the thrust distribution and the air skin friction on the blades influence its position, but the heart of the phenomenon is not seriously influenced by these considerations.

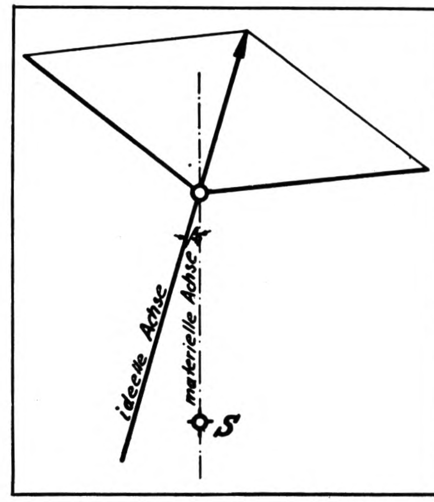

FIG. 11 DIAGRAM INDICATING ACTUAL AND VIRTUAL AXES OF ROTATION OF HELICOPTER ROTOR

If the center of gravity of the whole aircraft lies below the airscrews, it becomes evident that there is longitudinal static stability due to the airscrew alone. An increase in speed, that is, a greater inclination of the virtual axis allows the thrust to move ahead of the center of gravity, as β_1 increases. The converse is true with decrease in speed. As the same formulas serve for the helicopter as well as for the autogiro, the principle holds true for both types of aircraft. Therefore we can calculate the longitudinal moments in both cases, or determine them

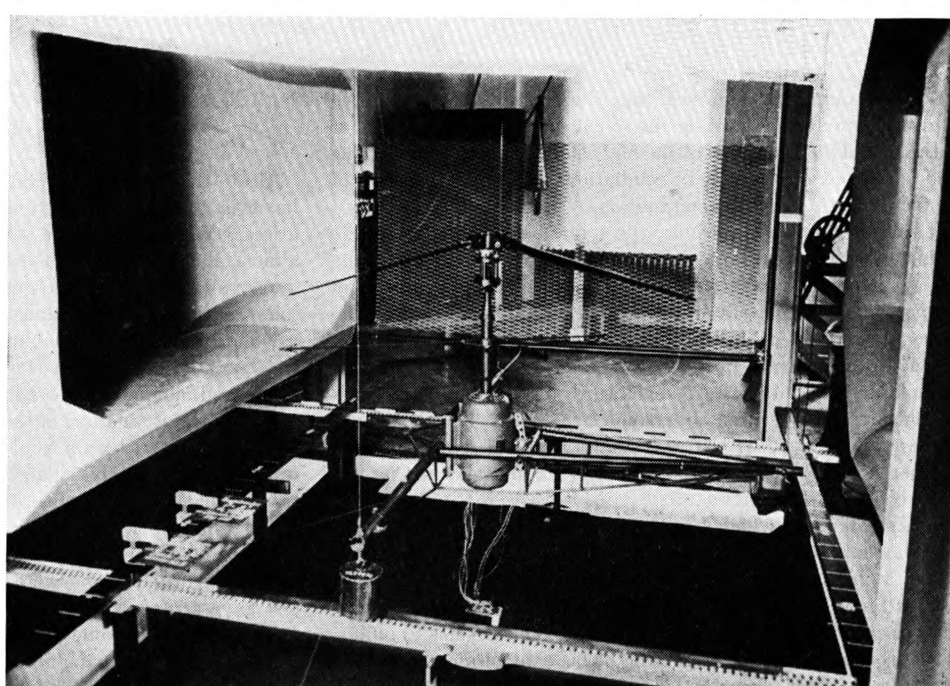

FIG. 12 ARRANGEMENT OF WIND-TUNNEL FOR STEREOPHOTOGRAPHIC MEASUREMENTS ON BLADES

by tunnel tests, in which both the magnitude and direction of the resultant can readily be obtained. As the resistance, because of many corrections and because of the size of the suspension wires, cannot be determined with great accuracy, we have been led to other experimental investigations.

First it was desired to know the position of the blades during the whole cycle of rotation, as many constructional requirements—for example the kinematics of the blade positioning—require such information. The English have carried out such an investigation with mirrors. To Professors Schering, Finsterwalder, and Flachsbart I owe the proposal and execution of a stereophotogrammetric method, which for an airscrew of $1^1/_2$ meters diam, gives the correct location of the blade at half radius with an accuracy of approximately 1 mm. See Fig. 12. Between the objectives of a stereocamera, an illuminating spark is introduced giving an illumination time of 0.000001 sec which, at a speed of 1700 rpm, allows us to see a row of white dots painted on the airscrew. With the airscrew and the wind tunnel operating, from 20 to 30 sparks are induced, giving on the exposed pair of plates a row of photographs of the white dots at random positions, but sufficiently well distributed to give a complete picture of all blade positions, after evaluation with the stereocomparator. An auxiliary system includes a measurement system on the floor of the wind tunnel. An example of such a photograph is shown

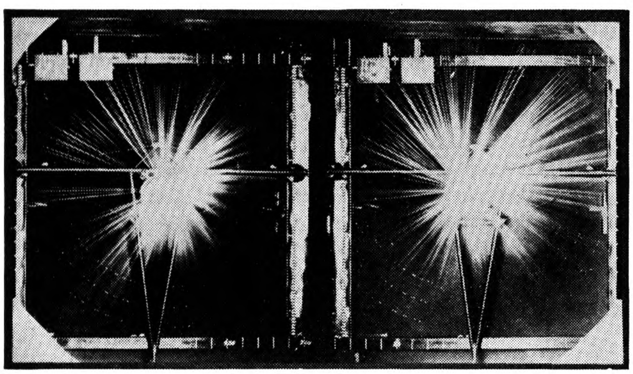

Fig. 13 Stereophotograph of Blades Revolving at 1700 Rpm
(Distance of white dots from the center is a measure of angle β.)

in Fig. 13. The star-shaped figure is caused by the reflections of the blades and has no significance. The method is well adapted to give information regarding the so-called change of phase, that is, the actual displacement of the highest and lowest points of the blade path out of the cone of revolution owing to the inertia of the blades during the flapping motion.

It follows that on the whole, the screw behaves not only like an airfoil with a constant center of pressure, but also like one having inherent stability. When this stability is produced by the position of the resultant, it becomes possible to control the rotary aircraft by deliberately changing the position of this resultant force. Cierva, in the C-30 autogiro, has secured control by inclining the hub forward, backward, or sidewise. Controllability has been secured, but some accidents when going into high speed, that is, with strong forces, lead to a doubt as to whether longitudinal stability is always maintained in Cierva's direct-control system.

In a small work which has been sent to interested parties, I have shown that a peculiar hazard is produced. The direction of control at high speeds changes signs or at least the control moment attained is independent of the positioning, so that the pilot has the sensation of loss of control. If the pilot decides to pull or push through even if the beginning of his control movement has the contrary effect, then the aircraft will eventually right itself. The real danger is that in the instant of apparent loss of control, such high diving speeds can be obtained that autorotation ceases. The thrust decreases, and the restoring longitudinal moment even after the successful effort, is insufficient to prevent further downward curvature of the flight path in space.

It is a mistake to think that the fixed stabilizer is the only guilty element. It was an error on Cierva's part to think that his direct control was solely a center-of-gravity control system. At high speeds, high moments are produced from the fixed stabilizing surface. I have therefore proposed that with rotary aircraft a movable horizontal surface should always be included which could work effectively under all circumstances to produce moment equilibrium. Then every danger on this score would be excluded. The D.V.L. has made independent investigations and has come to a similar conclusion. Nevertheless Cierva's direct control is not employable in the helicopter. It gives large control forces, which go into the whole aircraft, through the lever arm produced by β_1, which is not always negligible. If transmission is added to the system, then it will be seen that the large tooth and bearing loads, which change with the performance, cannot but have an effect on the control system when a tiltable hub is employed.

If, then, it is not possible to change the actual axis of the airscrew, we can change the "virtual" axis by varying β_1. We saw that the equilibrium of the air and dynamic forces prescribed the position of the flapping blades during a cycle of revolution, so that they took the corresponding angle of attack. As we must change the angle of the blades in going from helicopter to autorotative operation, we can reverse this process, and force the blades to have an angle of attack, variably controlled throughout the revolution, so that they must follow a desired path in space suitable for control.

Lock had already shown that the imposition of a sinusoidal variation in the angle of setting during each revolution produces the same effect as inclining the whole airscrew. Hence, control with the virtual or with the actual axis of rotation is equivalent, but it is important to note that control by means of varying the angle of setting, with the use of a blade having a constant center of pressure, theoretically introduces no forces into the control system. In practice, only the skin-friction forces and small mass forces are introduced. The change of the angle of attack is therefore an ideal control relay. Moreover the mechanism for the change of the angle of incidence is under little stress, and can be both light and reliable.

(b) *Dynamic Stability*

Up to this point everything has been relatively simple. Now, however, the most difficult question, that of dynamic stability, arises. When we began the construction of our machine, almost no information was available in this field. Nothing was known about either the helicopter or the autogiro rotor. We have followed two paths simultaneously. First we adapted the normal calculations for the airplane for use with the helicopter. Then we derived the equations for the hovering helicopter from the forces and moments involved. Numerical data were obtained in part by careful weighing of the degree of reliability of the Glauert theory, and in part from wind-tunnel work. Particular attention was paid to the stability and control processes at the instant of transformation from helicopter to autogiro flight. From these calculations we were able to give the pilot careful instructions about the movement of the controls and the setting of the stabilizer. The calculations were verified in practice, and the first test gave a perfect three-point gliding landing from a height of 400 m (1300 ft). Some two seconds after the transformation, the machine was in normal gliding flight. One can say that with this achievement of Pilot Rohlfs on May 10, 1937, the realization of the practical helicopter was assured. The haunting thought of power-plant failure had lost its terrors.

Of our fundamental requirements we have therefore obtained both stability and calculable controllability about all three axes, both for hovering and in forward flight. We have obtained the transformation from helicopter into autogiro glide flight in a few seconds. We have secured simplicity and reliability in the arrangement of the control system, since with only two principal movements on the screw hub, there have followed four controls, two linear—transformation and rolling control—and two of a sinusoidal character—height and yawing control. Finally, the condition of simple effort by the pilot has been met because the three customary movements of hand and feet have only been increased by another operation with a lever placed at the right side of the seat. This is for the transformation from helicopter to autogiro rotor, and is used only in take-off and in special emergency cases.

I cannot conceal the fact that the theoretical and experimental effort in securing these results was very great. In particular, the stability calculations were most complicated. As one is entering unknown territory it is necessary to be very conscientious and to avoid all approximations and neglecting of terms. Where approximations could not be avoided, due to mathematical limitations, they had to be supported by special lengthy investigations. The success attained has justified this enormous effort. The first free flight of the machine lasted 28 sec, the fourth 16 min, and, although much is to be attributed to the skill of the pilot, Rohlfs, such success in the tests would have been unthinkable without a thorough technical understanding of the entire subject.

Performance

(a) Autogiro

In considering the polar of the autogiro rotor, Fig. 3, it is striking that the drag is large with large values of C_a, many times greater than that of a good airfoil. That is explainable not only on the ground of the low aspect ratio $\pi D^2/4D^2 = \pi/4$ which is unchangeable, but also is traceable to the C_{wp} which is a special characteristic of the autogiro rotor. On the other hand, with small values of the lift coefficient C_a, C_w is small, that is, the drag coefficient is small at high speed. This circumstance is specially emphasized by the small loading of the blades when referred to the whole circular disk. The lift-drag ratios at small lift coefficients approach those of good airfoils, and lie around 8 to 1 or 10 to 1. Even an L/D of 12 seems attainable. But these are, in contradistinction to the airfoil, the highest possible values of L/D. At high speeds therefore the autogiro rotor should not be far inferior to the fixed airfoil. The autogiro will have a maximum speed about 15 per cent less than that of the airplane.

Schrenk has given an interesting comparison between the airplane and the autogiro in the form of characteristics curves, Fig. 14. The really striking difference rests in part on the greater parasite drag, which difference could be reduced by better streamlining of the hub, starting rig, and other parts. Therefore it can be estimated that the autogiro will be some 10 per cent lower in top speed than the airplane but, on the other hand, will have about half the minimum speed.

Things look worse in the climb. Now we are working on the higher parts of the polar curve and the resistances become large. The lowest power required is much higher than with the airplane and also occurs at slower speeds, which is undesirable because of the resultant inefficiency of the propeller. That strengthens us in the conviction that the autogiro is only destined to play an intermediate part between the airplane and the helicopter. The autogiro solves less than half the problem, that of landing in a small space, but does not permit a zero velocity, while the take-off and the performance in climb are worse than before. There is, however, no inferiority from the point of view of weight. On the contrary, in large machines the freedom from bending moments should more than counterbalance the weight of the hub and starting gear. This is one of the great advantages of all rotary aircraft.

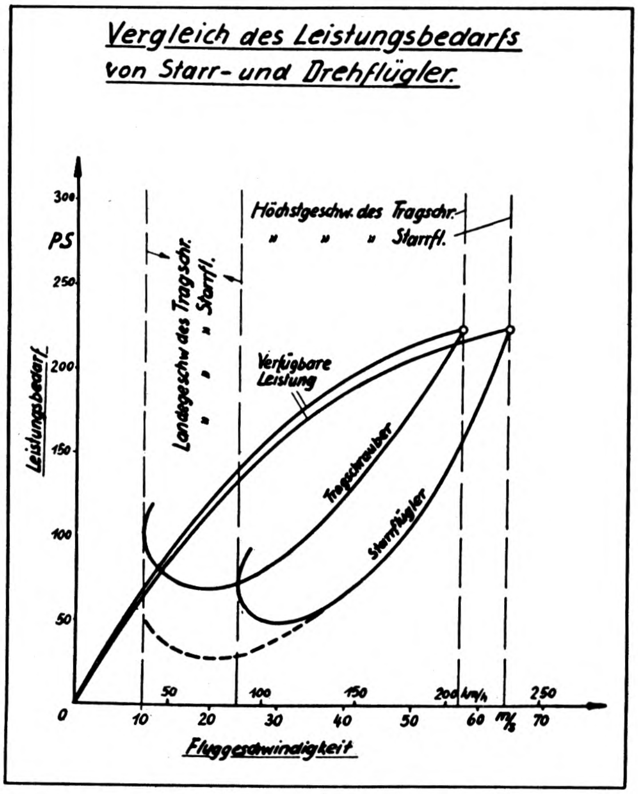

Fig. 14 Comparison of Power Requirements of the Airplane and Autogiro

(Ordinates, power required, hp; abscissas, flying speed, m per sec and km per hr; landing speed of autogiro, 10 m per sec; landing speed of airplane, 25 m per sec; maximum speed of autogiro, 57 m per sec; maximum speed of airplane, 65 m per sec; *Verfügbare Leistung* = *power available*.)

(b) Helicopter

We must make proportionately the largest sacrifice regarding weight in the helicopter. The transmission must function with complete reliability under full output of the motors, which places the helicopter in small sizes at a serious disadvantage with the airplane. Bréguet has on the other hand designed a 16-ton helicopter flying boat, and finds a saving in weight as compared to the corresponding high-speed airplane. If this appears somewhat optimistic the fact nevertheless remains that, with greater size, the weight of the transmission of the helicopter will become less of a disadvantage.

When high speeds are considered, the question arises: Is it advantageous to let a given rotary aircraft fly as an autogiro or as a helicopter without a propulsive airscrew? It is conceivable that a machine could be designed to have zero velocity and yet be flown as an autogiro. We have made careful computations, reinforced by wind-tunnel tests, with the interesting result that the same machine flown as a pure helicopter is somewhat faster, even if we deduct 10 per cent for motor cooling. Practical experience with a helicopter has fully confirmed this view. The machine after installation of a normal propeller instead of the cooling blower could be flown as an autogiro, and gave with streamlining of all tubing and parts a speed of 120 km (75 miles) per hr, as compared with the 126 km (78 miles) which had been calculated. Streamlining for the helicopter could not be incorporated without reconstruction. The horsepower requirements of the unstreamlined bracing and parts could be determined in special tests, however, and at 120 km (75 miles) per hr, amounted to 46 hp. Thus the streamlined helicopter would be 25 km (15 miles) kilometers faster. That would give 147 km (91 miles) an hr for the helicopter as compared with the calculated value of 152 km (94 miles) per hr.

As we saw that the autogiro, in so far as top speed is concerned, is only slightly inferior to the airplane, we can conclude that the helicopter could at least equal the speed of the airplane. Here also the decrease of the parasite resistance is the chief point of further development. There is, however, a certain limitation to the helicopter. As we cannot obtain a forward-speed parameter of over 0.5, the tips of the helicopter blades will reach the speed of sound at half the forward speed of the airplane. Then the addition of speed has to be reckoned with in the advancing blade. At a speed of about 400 km (250 miles) per hr, the prospects of the helicopter become much worse than those of the airplane. Of course it must be remembered that in the same speed ranges, difficulties regarding compressibility effects also begin to occur with the conventional propeller.

The climb of the helicopter appears to be remarkable. In Table 1 are shown the initial climbs of the Fw 44 Stieglitz, of the de la Cierva C 30, and of the helicopter, flown as an autogiro and flown as a helicopter, all of them provided with the same motor, the Sh 14. In particular it should be noticed that there was no equality of gross weight. The helicopter is the heaviest, nevertheless it has the greatest climb. The autogiros are quite inferior.

TABLE 1 INITIAL CLIMBS AND WEIGHTS OF VARIOUS TYPES OF AIRCRAFT WITH THE SAME MOTOR

Machine	Initial climb M per sec	Initial climb Fps	Flying weight Kg	Flying weight Lb
Fw 44 Stieglitz airplane	3.5	11.5	870	1920
Cierva C30 autogiro	1.5	4.9	815	1795
Fw 61 helicopter as an autogiro	1.3	4.26	950	2095
Fw 61 helicopter as a helicopter	3.6	11.8	950	2095

I must particularly emphasize here that the helicopter does not secure its justification by the existence of superior climb or speed, relative to those of the airplane. Its special characteristics give the helicopter special duties and possibilities. It will be all the more worth while however if it shows that it is not inferior in performance characteristics.

It is hardly necessary to speak of the take-off and landing characteristics. The ideal sought for is attained, at least technically. Doubtless the full utilization of all the possibilities of the helicopter will make new demands on the skill of the pilot and the personal equation will enter. But I can report that contrary to anticipation, no acrobatic skill is needed. Miss Hanna

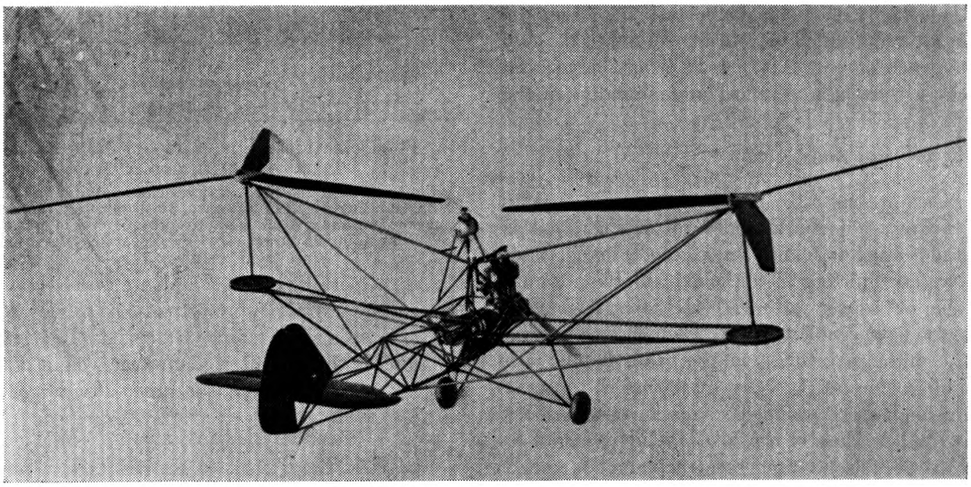

FIG. 15 FREE-FLYING MODEL OF HELICOPTER

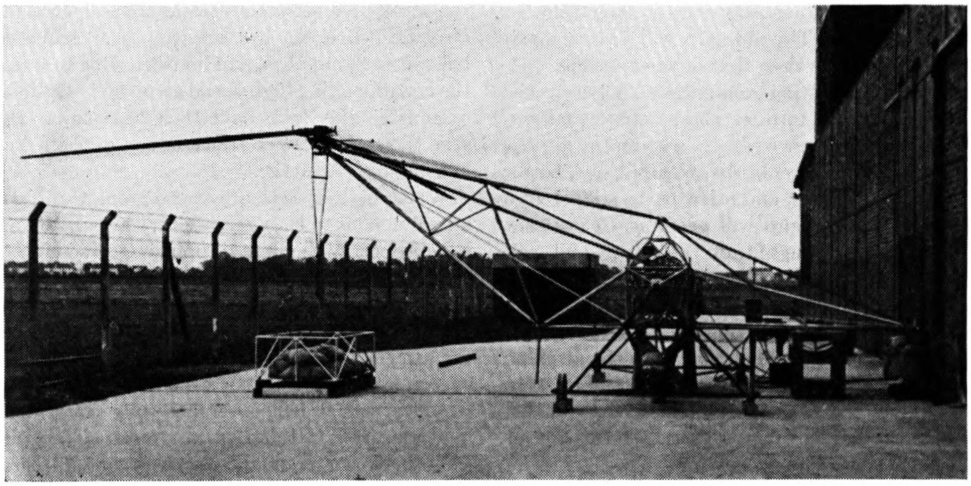

FIG. 16 HELICOPTER WITH ONE ROTOR LOADED FOR TESTING

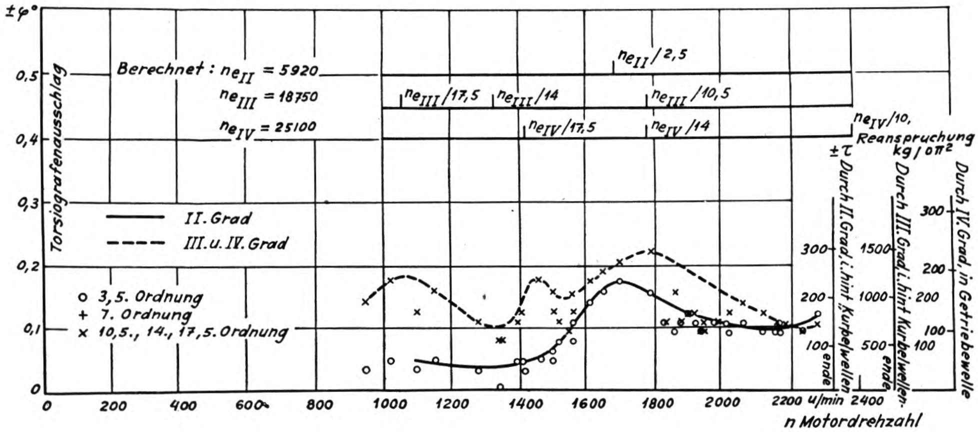

Fig. 17 Torsiogram of Power Transmission

Reitsch and Messrs. Dipl. Ing. Francke, Ballerstedt, and Bode have flown my helicopter splendidly with short technical instruction, and made smooth landings.

Due to the accidents with autogiros caused by cessation of autorotation at small angles of attack and large pitch angle, and to the consideration that small angles of attack must inevitably occur in gusts, the opinion has arisen that all rotary aircraft are sensitive to gusts. As the connection with autorotation shows, this conclusion is not at all applicable to the helicopter, and we have seen already that with the autogiro there is no danger if an adjustable horizontal surface is provided. Besides it is to be surmised that, as a result of the flexibility of the blades, the rotor would be less sensitive to gusts. By means of simultaneous tests of an airplane and of the helicopter, it was shown that the pilot of the airplane experienced severe bumps, while the pilot of the helicopter hardly felt the gusts. At other times the gustiness was perceptible at wind velocities of 40 km (25 miles) per hr. It therefore seems probable that the character of the gust is to be considered. It is interesting to learn that, on the small factory landing field of Focke, Achgelis & Co., the helicopter was easily controllable, even in the disturbed regions of buildings, in gusts of from 6 to 10 m (20 to 33 ft) per sec.

I have so far considered only the investigations which were necessary to the development of the helicopter. I cannot emphasize too strongly in what measure pure science was the foundation on which new knowledge was based. The second and still more difficult stage of our work was in embodying in practice the information thus gained.

Construction

The first step was a free-flying model, driven by a 0.7 hp two-cylinder motor. See Fig. 15. It had a flying weight, with 50 g of benzene, of 4.9 kg (10.8 lb). It is easily understandable that the model was more frequently in bits than whole, nevertheless many worth-while experiments were made with it. In November, 1934, it reached a height of 18 m (59 ft), equaling what was up to that time the world's record for a helicopter with a pilot. The model is now in the Deutsches Museum in Munich.

It was decided further that the transmission, coupling, and flap control should be submitted to the same tests as a new engine. The Brandeburg Motor Werke, which under the personal direction of Director Wolff, had undertaken the difficult task of this construction and of the conversion of the Sh 14-a motors, built one side of the system. See Fig. 16. It was provided with a single airscrew and its structural supports. A Leonhard electric drive was used, so that the performance requirements of the airscrew could be measured in full scale. The thrust was measured by means of ballast and by making the fuselage rotatable about its longitudinal axis, taking due account of the ground effect. This constituted a valuable supplement to the wind-tunnel tests at full-scale evaluation. A 50-hr test was made and, after overhaul of the transmission, another 10-hr test. The controls were continuously tested at the same time.

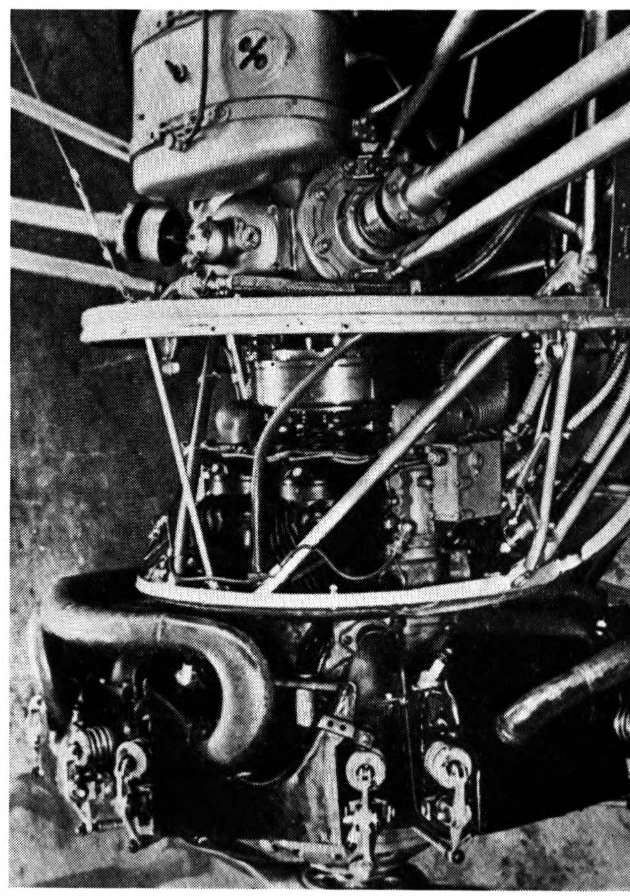

Fig. 18 Showing Engine, Friction Clutch, and Automatic Mechanism for Changing to Autorotation

A vibration investigation of the whole transmission system was made for the D.V.L. by Professor Lürenbaum and his associates. The calculations were confirmed by tests with the torsiograph, in which the aircraft was flown as a kite. This was all the more necessary since the electrical drive used in the first test was not

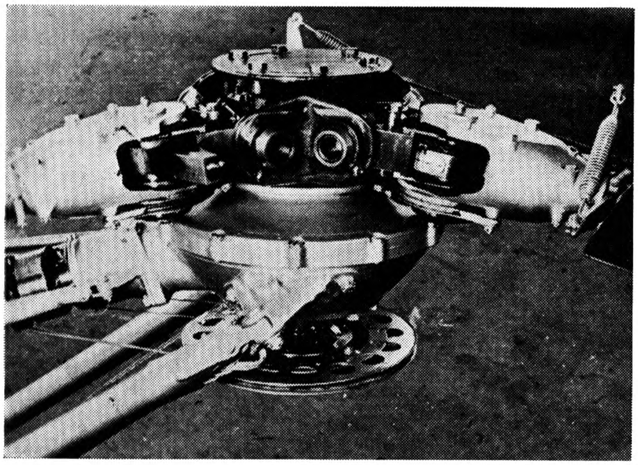

Fig. 19 Rotor Hub With Blade-Pitch Controls

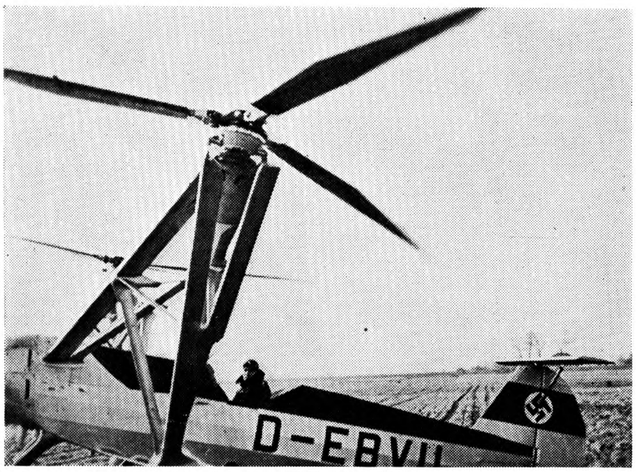

Fig. 22 Pilot Rohlfs and the Machine After Breaking All World Records

Fig. 20 Showing Propeller-Type Blower for Cooling Engine in Hovering Flight

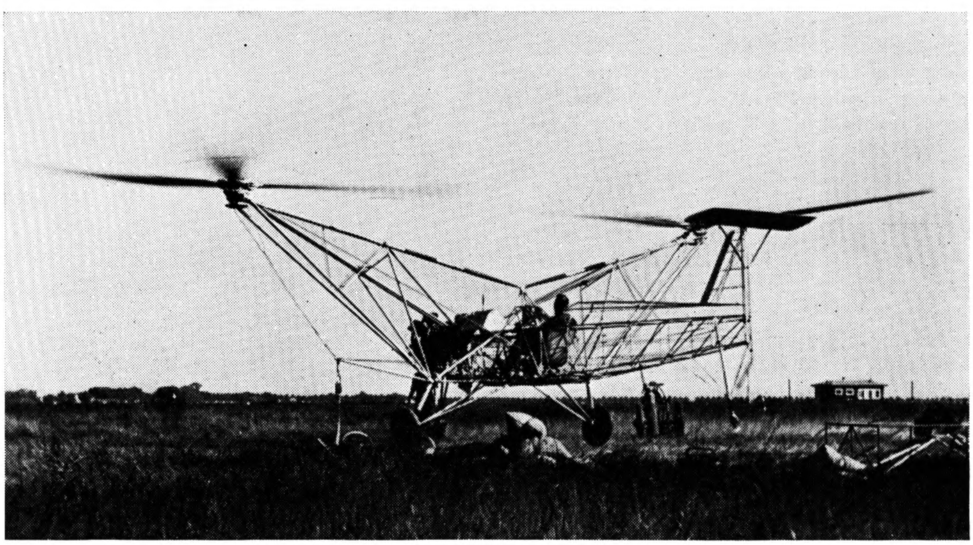

Fig. 21 Completed Helicopter in Captive Flight

sound from a vibration point of view. Fig. 17 shows a torsiogram, of which I will only say that the most highly stressed part was within the motor and not within the helicopter. Fig. 18 shows the transmission with the friction coupling, and the safety provisions which, by failure of the motor or of the gears, or the reaching of a certain minimum rotating speed, automatically causes transformation into the autogiro state. Fig. 19 shows one of the screw hubs with controls for the blade motion.

The construction of the framework and of the rotors was undertaken as for a normal airplane. New ideas in the design office and the unconventional constructional elements made difficult work for my associate Ing. Körper. Under these conditions it is gratifying that maintenance, while not ideal, is adequate. For the new problems in stress analysis, the calculations and experiments previously made were utilized, and in some respects these had to be supplemented.

For the cooling of the motor in flight in one spot, a propeller-type blower was developed, and the cooling was investigated by cylinder-temperature measurements in cooperation with the Brandeburg Motor Werke. See Fig. 20.

Flights

The completed prototype was often flown in a captive condition, as shown in Fig. 21. Captive flights are an excellent method of investigation, as everything happens under flight conditions while one is only from $1/2$ to 1 meter above the ground. No further steps in the proving were undertaken until the conditions were clarified by further calculations or tests. On June 26, 1936, Rohlfs made the first flight. On June 25 and 26, 1937, he was able to bring to Germany all the world helicopter records, with fifteen times the performance existing up to that time. See Fig. 22. I have already spoken of the important glide landings. The height reached, 2439 m (8000 ft), which did not long remain the maximum height, has brought me the undisguised charge of deceit from Mr. Asboth in a foreign technical paper. The lack of foundation for this accusation is shown by the many witnesses of the flight and through the immediate recognition of all records by the Fédération Aéronautique Internationale.

In June and October, 1937, the first and second machines, respectively, were taken over by the Reich. With the latter Miss Reitsch flew, in October, 1937, from Bremen to Berlin surpassing the record between Stendal and Tempelhof by over 108 km. Bode, on June 20, 1938, surpassed this record by flying 230 km after a larger fuel tank had been installed.

Nobody had expected such performances. The future is bright.

The Interpretation of a Failure of an Ordnance Structure[1]

By G. F. Jenks,[2] Washington, D. C.

For the analysis of a failure of a war model of a 240-mm howitzer at half pressure, a fragment was furnished the laboratory. The problem was divided into three phases: First, the origin of failure; second, the mechanism by which the failure developed in the structure; third, the conditions responsible for the failure. Emphasis is placed on determining the spot at which the failure started.

A method of reading the rupture flow lines found on the surface of the fragments is described. The arrow points found in torn metal surfaces are shown as pointing to the origin of rupture. These arrow points and other surface characteristics furnish the time scale by which priority as to time may be established for tears or for ruptures which appear to be independent. Once the beginning of a failure is found, however, the laboratory investigation then becomes simple as it may be confined to a small area.

In this investigation the heat-affected zone of a weld was disclosed by acid etching. The design called for no welding at this point. Microscopic investigation showed that weld metal of an inferior quality was present, and that cracks existed in the heat-affected zone of the weld. A strip of metal of high Brinell hardness was found just outside the weld metal. The failure progressed from old cracks in this region of high hardness on the exterior surface of the gun. The tension and impact properties and the macroetch characteristics of the metal were determined and their influence upon the failure estimated. Finally, manufacturing records were checked and showed that a machine error on the surface in question had been repaired by welding.

The design of a structure is based upon an analysis of applied stresses with provision for errors in the method of calculation and for deficiencies in behavior of materials applied under the factor of safety. Failures of structure may occur for any of the following reasons: (1) Stresses applied in excess of those for which the structure was designed; (2) highly localized stresses; and (3) unpredicted behavior of the material. Designs may be checked by various proof tests but primarily by service use. If structures fail under service conditions a complete analysis of the failure is essential to the perfection of design and processing practices.

In the study of failures of structures the first task is to find the origin of the failure; the second, the mechanism by which the failure developed in the structure. After this preliminary work the examination of the material and the analysis of stresses point to the conditions responsible for the failure.

Each structural failure is a special problem and involves different lines of approach. It is believed, however, that cases of examination of failed structures are of interest. Engineering literature is comparatively poor in this respect. There is a need for a more general disclosure of information regarding failures and their analysis for the improvement of design practice.

The gun is one of the earlier engineering structures for which careful stress analysis was made. It differs from most engineering structures in that the factor of safety is low and in that its life is limited by erosion of the bore rather than by the fatigue strength of the metal. This paper deals with the analysis of a failure which has taken place in a 240-mm howitzer of war production.

Howitzer No. 5 ruptured on the 283rd round on a half charge with a bore pressure of 15,400 lb per sq in. at the breech. The howitzer is designed for a bore pressure of 33,000 lb per sq in. with a minimum factor of safety of 1.25 in the powder chamber. The factor of safety of the muzzle section was not much over unity. Expansion of this section had been experienced under service pressures until a change in the contour of rifling was made.

Fig. 1 shows the gun and mount immediately after the failure. The fragment not shown was found 155 ft to the right rear. The muzzle was split in two sections, one of which is shown in Fig. 2. The two ruptured surfaces of this section are quite different in appearance. The lower surface is smooth in the middle region and fibrous at the ends, the coarseness of the fiber increasing toward the end. The top fracture has no smooth area except two very limited ones near the bore at *B* and *C*. The ruptures are characterized by fibers torn in the general shape of an arrowhead. In the lower surface the points of the arrows are directed toward the smooth area at the middle section. The fibers in the upper surface are coarser and the surface is rougher. The arrows of the fiber point toward *C* and also in the left section toward *B*. Fig. 3 shows the middle portion of the lower fractured surface. It will be noted that just below the center of the illustration is a very smooth area and that the apex of the arrows of the fibers point to that area. The lower edge at the picture is the outside surface of the gun. In Fig. 4 is shown one of the smooth areas of the upper ruptured surfaces of Fig. 2. The arrows of the ruptured fibers point to a small area at the lower (bore) surface of the fragment.

From general observation it has been noted that the apex of the arrows in a fractured surface point toward the origin of rupture. Fig. 2 shows three origins of rupture, one in the lower surface near the outside and two on the upper surface near the bore.

To establish priority of these origins, the nature of the surfaces must be considered. The lower surface of Fig. 2 (see also Fig. 3) is smoother and the center of propagation of rupture extends along the mid-section of the wall. There is a single system of rupture lines from one center. The upper surface of Fig. 2 (see also Fig. 4) shows the path of rupture extending along the bore surface. Fig. 4 shows that the flow lines change in appearance before reaching the upper (outside) surface. The rupture

[1] Released for publication by the Chief of Ordnance, U. S. Army. Statements and opinions are to be understood as individual expressions of their author and not those of the Ordnance Department.

[2] Colonel, Ordnance Department, U. S. Army. Presented at the Annual Meeting of The American Society of Mechanical Engineers, held in New York, N. Y., December 5–9, 1938.

Discussion of this paper was closed January 10, 1939.

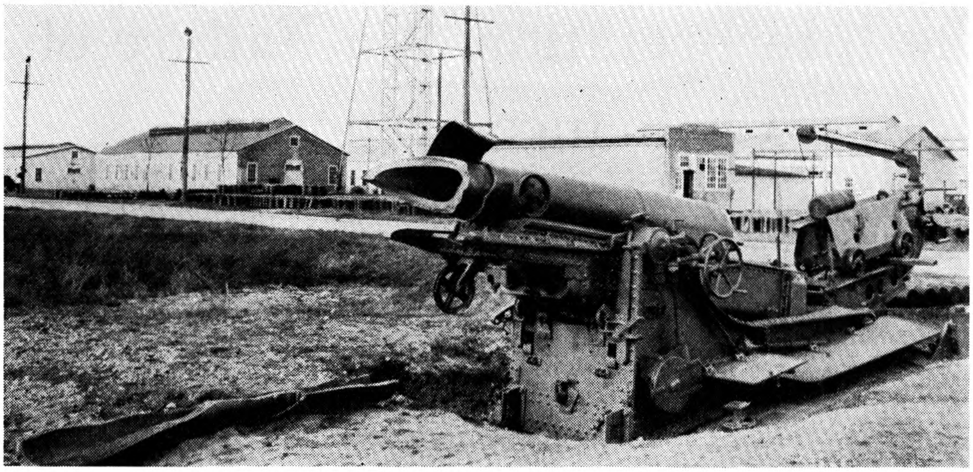

FIG. 1 FAILURE OF 240-MM HOWITZER

lines of the lower zone formed by about two thirds of width of section are traces of tearing. The upper zone has the appearance of having been pulled apart rather than torn. Near the upper edge of the upper fracture shown in Fig. 2 there is a difference in level of the fracture indicative of forces which did not exist in the rupture of the lower surface. There are at least two systems of rupture lines in this surface.

The smoothness of the ruptured surface varies directly with the speed of propagation of rupture.

The only conclusion that can be drawn is that the original rupture started near the outside surface at the point marked "start of rupture" in Figs. 2 and 3, proceeding in both directions. With the split on one side of the gun, the unsupported cylinder rotated about the outside edge of the opposite wall, causing failure to proceed from the inside surface outward. The initial fracture was more rapid. As the powder pressure was reduced, with loss of gases through the split and the increased volume of the bore, the violence of rupture decreased. Thus, the upper surface in Fig. 2 is rougher in appearance. As the two halves of gun opened out on the upper edge of Fig. 2 at a center, the rupture in this wall changed from a tear to a tensile break, as indicated in Fig. 4. In the second surface ruptured, the presence of two or more origins is natural. Both were due to stresses beyond the strength of the material induced by the failure of the opposite wall.

The appearance of the ruptured surfaces is consistent with hypothesis as to the origin of fracture. The position of fragments on the ground after the accident is also consistent with this hypothesis. The rupture flow lines shown in Fig. 1 definitely indicate the tearing proceeding from the near side, which checks with the hypothesis made from the examination of the fragment shown in Fig. 2.

A closer examination of Fig. 3 discloses a smooth, dark zone at the outer surface about $^1/_8$ in. deep and 4 in. long. An examination of this surface at low magnification, which is shown in Fig. 5, reveals a number of smooth dark areas which are interpreted to be independent crack systems of varying depth, the principal ones being near the two ends of the illustration. Near the middle of the upper edge will be noted a center from which rupture flow lines proceed in all directions and merge into the main rupture flow system. This is the localized origin of rupture at the bottom of an old crack system. It should be noted that this rupture system comes to the surface in some places and is indicative of the discontinuity and variable depth of the original crack system. Particularly to the left of the origin of rupture

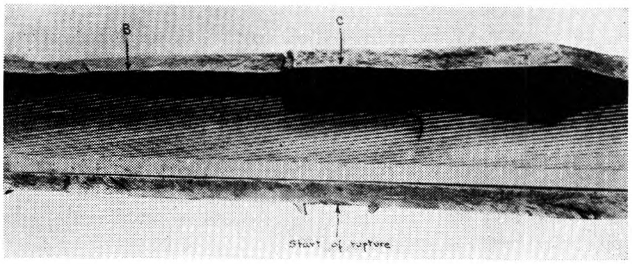

FIG. 2 ONE SECTION OF THE SPLIT MUZZLE

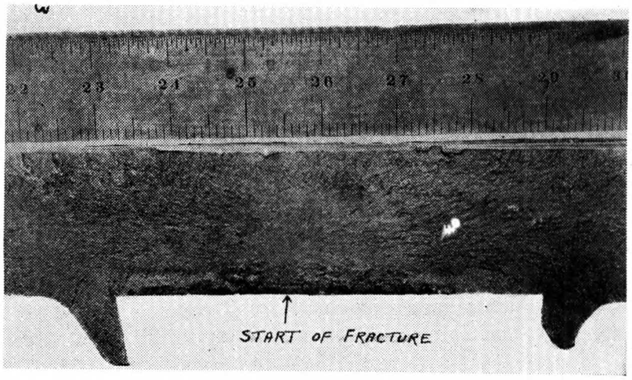

FIG. 3 MIDDLE PORTION OF THE LOWER FRACTURED SURFACE OF THE MUZZLE

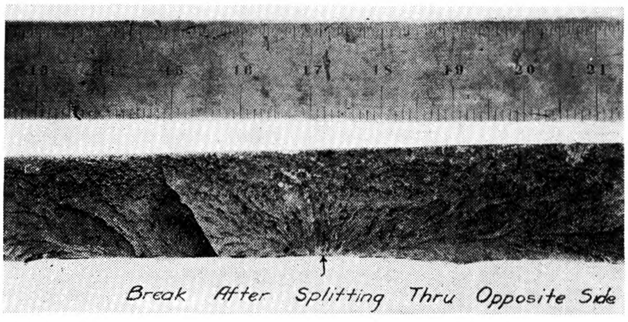

FIG. 4 ONE OF THE SMOOTH AREAS OF THE UPPER RUPTURED SURFACES SHOWN IN FIG. 2

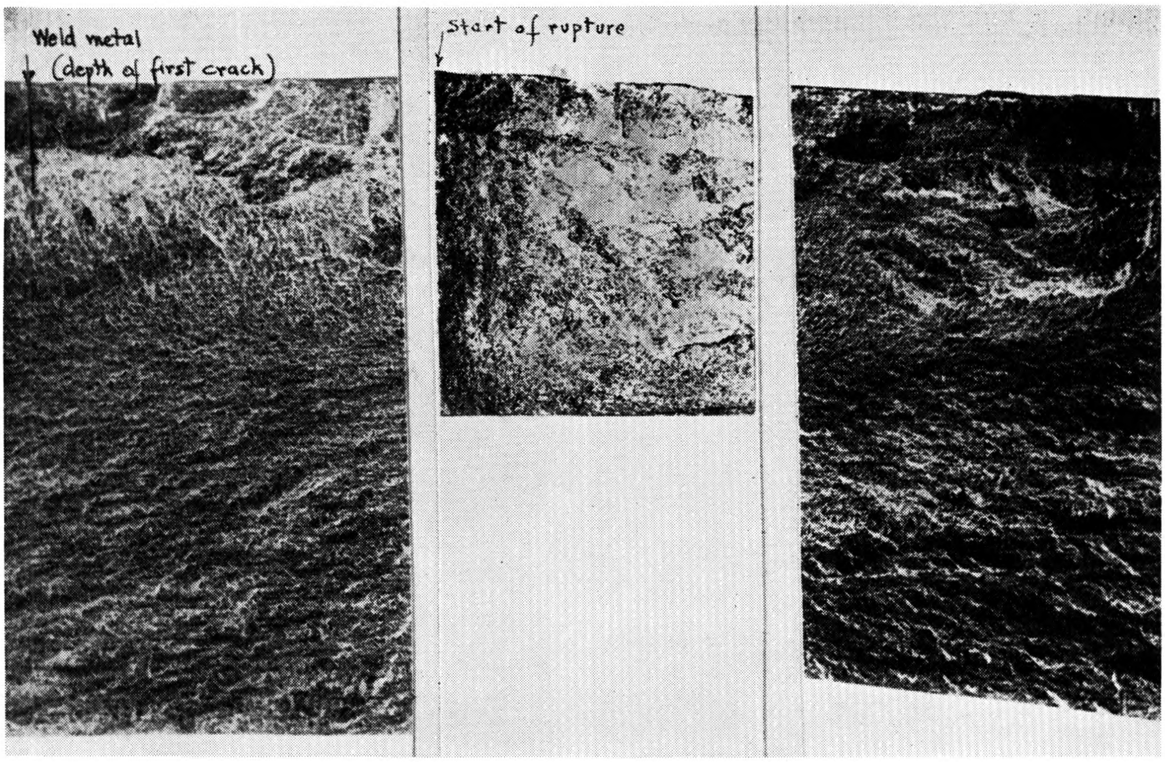

FIG. 5 LOW MAGNIFICATION OF THE SURFACE SHOWN IN FIG. 3

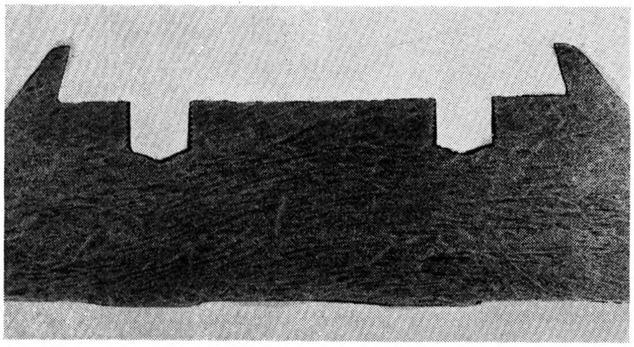

FIG. 6 MACROETCHED LONGITUDINAL SURFACE

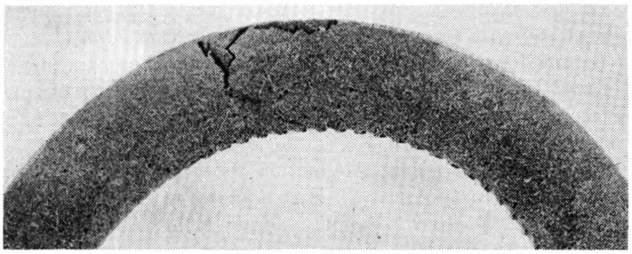

FIG. 7 MACROETCHED TRANSVERSE SURFACE

there will be observed independent flow-line systems. However, during the failure these local flow-line systems proceeded only a short distance before they met the main rupture system which was being propagated at a very high rate of speed.

Having definitely established the point from which the rupture proceeded, the next phase of the investigation is to determine under what conditions the original crack system developed and to examine the metal to ascertain why it has behaved so unexpectedly. Merely classifying the failure as one in fatigue gives no real answer.

Fourteen test specimens taken from metal of the fragments gave the results shown in Table 1, which are compared with original acceptance tests.

The lowness of proportional limits is explained in part by the cold work received by the metal in rupture and possibly in part to the use of more exact methods of test during the investigation. The difference in tensile strength indicates a difference due probably to location of specimens. The higher values for specimens originally tested may have been due to end effect in heat-treatment. That is, specimens near the end of the forging were hardened more in the quenching operation.

The average tensile impact strength is good; the low value is unsatisfactory. Two specimens gave fair values. Otherwise, the results were good. The ruptured surfaces of the tension-test specimens were characterized as laminated, with streaks of lustrous segregations and numerous checks on stems. These types of fracture and irregularity of tension impact tests are indicative of nonhomogeneity of structure.

The macroetched structure, illustrated in Fig. 6 for longitudinal surface and Fig. 7 for transverse surface, confirms the conclusions made from the appearance of fractures and the re-

TABLE 1 TEST RESULTS FROM FRAGMENTS OF THE RUPTURED MUZZLE COMPARED WITH ORIGINAL ACCEPTANCE TESTS

	Fragments	Original
Proportional limit, lb per sq in.[a]	37000–50000	63000–71000
Average	46700	....
Tensile strength, lb per sq in.	90000–93000	95000–101500
Average	91800	....
Elongation, per cent	20.5– 24.5	20.5– 22.0
Average	22.7	....
Reduction of area, per cent	38.7– 49.6	37.3– 49.2
Average	45.3	....
Tensile Charpy, ft-lb	10.7– 29.4	Not tested
Average	23.6	....

[a] One half of the specimen showed no proportional limit, the stress-strain line being a continuous curve.
NOTE: All specimens were transverse.

Fig. 8 Dendritic Structure Near the Bore Observed After Etching With Copper Chloride. ×5

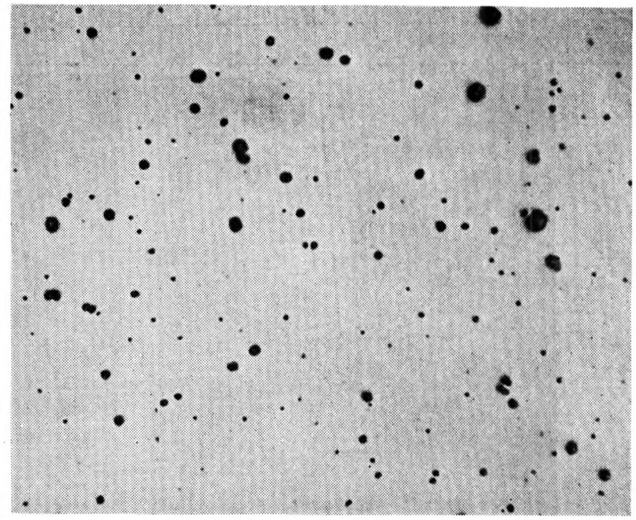

Fig. 9 Unetched Transverse Section Showing the Size and Distribution of Nonmetallics. ×100

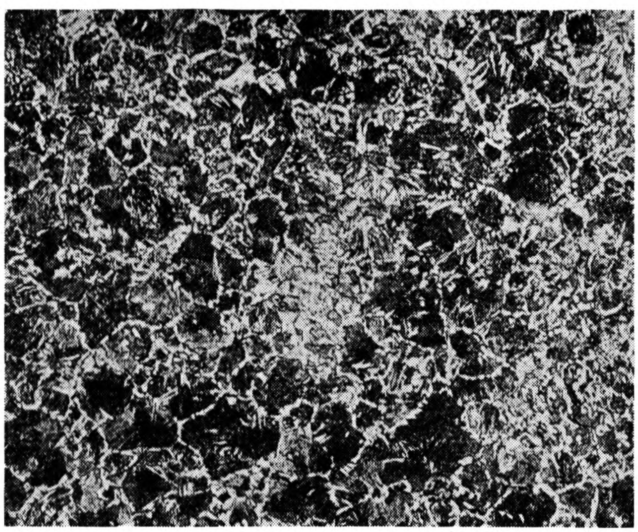

Fig. 10 The Structure After Etching With 1 Per Cent Nital. ×100

(Normal structure of the steel is a segregated nonuniform sorbite of variable grain size. Ferrite surrounds the grains and occurs in crystallographic planes.)

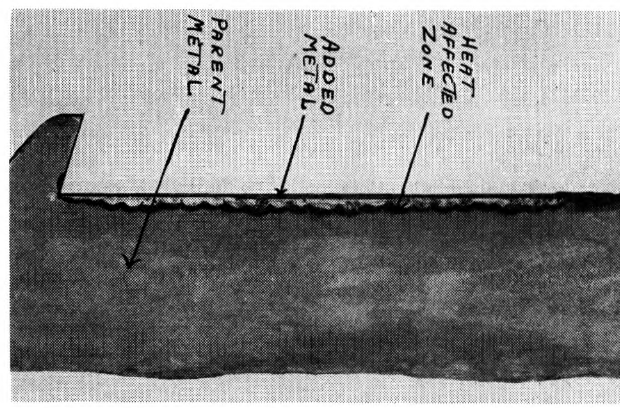

Fig. 11 Section Parallel to the Rupture and About 1 In. From It, After Etching With Nitric Acid

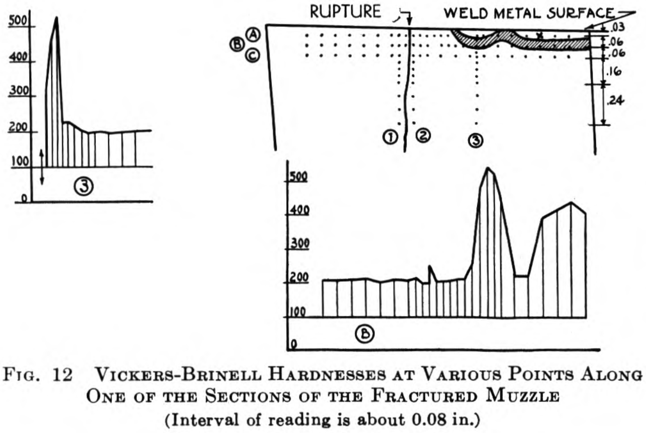

Fig. 12 Vickers-Brinell Hardnesses at Various Points Along One of the Sections of the Fractured Muzzle

(Interval of reading is about 0.08 in.)

sults of physical tests. The metal was of the following percentage composition: Carbon, 0.38; manganese, 0.64; silicon, 0.19; sulphur, 0.045; phosphorus, 0.061; nickel, 2.24; and chromium, 0.11.

The structure indicates inadequate hot working to break up the coarse dendritic structure of the ingot and to disseminate the phosphorus and sulphur content which was high. The structures found are favorable for propagation of crack systems. No evidence was noted, however, to indicate that any original crack system existed because of conditions of nonhomogeneity of composition or structure.

Under metallographic examination the pronounced presence of dendritic structure near the bore is shown in Fig. 8 at a magnification of 5 after a copper-chloride etch. An unpolished specimen, shown in Fig. 9, reveals the size and distribution of nonmetallics. Fig. 10 shows the structure after etching with 1 per cent Nital, which may be classed as nonuniform sorbite of variable grain size. It will be noted that the grains are surrounded by a network of ferrite and that ferrite also occurs in the crystallographic planes. The structure found correlates with irregularities of ductility and impact properties and the appearance of fractures of tension specimen. Low ductility and an increased rate of propagation of failure of the structure might be postulated from these results. As yet, however, the results do not explain the failure.

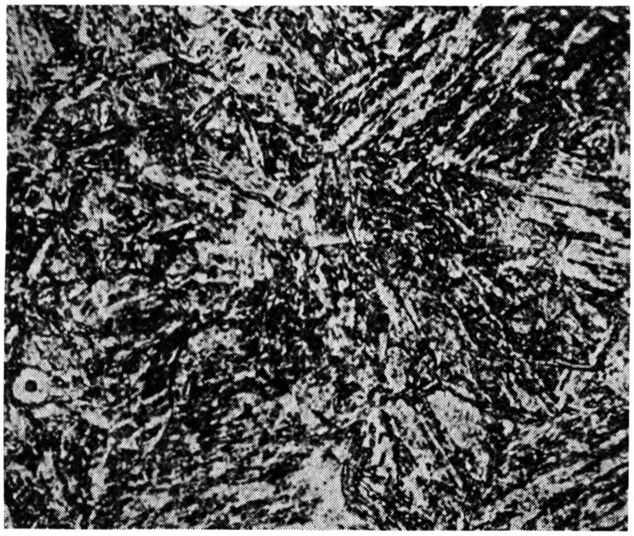

FIG. 13 SECTION NEAR THE FRACTURE ETCHED WITH 1 PER CENT NITAL. ×1000

(The structure has the acicular planes of martensite, but has been tempered by subsequent welding to troostite.)

FIG. 15 VOIDS IN THE WELD METAL WHICH WERE ALSO THE STARTING POINT OF CRACKS. SECTION ETCHED WITH 1 PER CENT NITAL. ×25

(The path of rupture did not pass through the added metal at the upper right, but through the heat-affected zone in the lower section.)

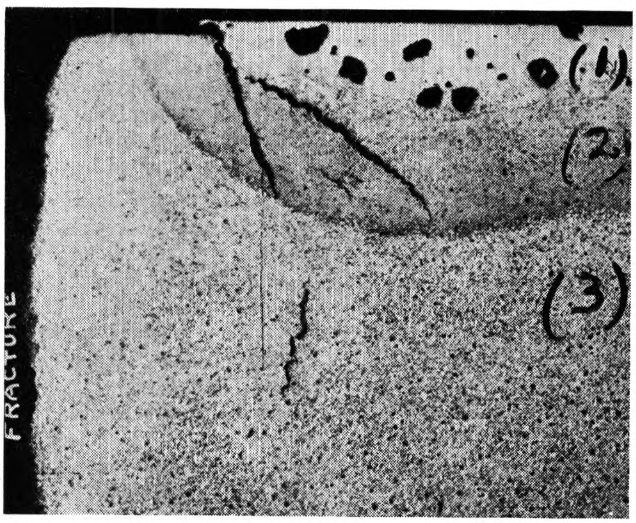

FIG. 14 PORTION OF THE WELD IN AN AREA NOT FAR FROM THE ORIGIN OF RUPTURE, AFTER ETCHING WITH 1 PER CENT NITAL. ×15

(Area 1 is the weld metal, area 2 is the heat-affected zone, and area 3 is the parent metal.)

A section parallel to the rupture and about 1 in. from it was macroetched and is shown in Fig. 11. This section reveals a weld on the surface in which the rupture had its origin. A hardness survey disclosed hardnesses up to 550 Vickers Brinell in the heat-affected area. Fig. 12 shows two hardness plots—*B* taken parallel to the weld and 3 from the edge of the weld through the heat-affected zone into the parent metal. The microstructure in the heat-affected area, shown in Fig. 13, reveals acicular planes of martensite tempered by subsequent welding to troostite.

A microstudy of the weld in an area not far from the origin of rupture was made. In Fig. 14 the weld deposit is shown at (1), the heat-affected zone at (2), and the parent metal at (3). Of particular interest is the shoulder at the left edge of the weld, the crack system starting from this shoulder, and the intergranular-crack system in the parent metal.

Other intercrystalline cracks were found in the heat-affected zone near the fusion line. In some areas, weld undercuts were the starting point for intergranular cracks. Voids in the weld metal were also the starting point of cracks, as indicated in Fig. 16 which shows the weld metal at the upper right and the heat-affected zone in the lower section. In some areas cracks were transcrystalline. Most of the intercrystalline cracks are believed to have existed prior to rupture.

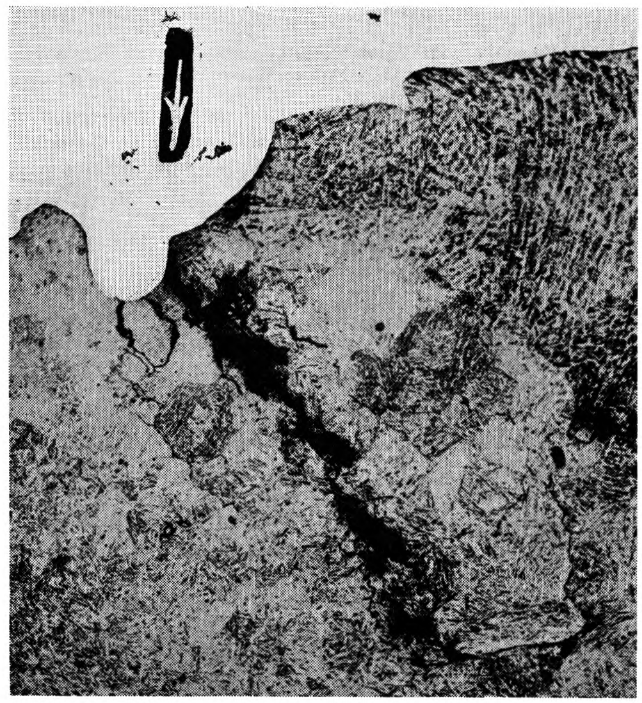

FIG. 16 ANOTHER TYPE OF DISCONTINUITY FOUND IN THE FUSION ZONE. SECTION ETCHED WITH 1 PER CENT NITAL. ×100

(The arrow points to undercut. The weld metal is in the upper right-hand corner. The crack system originated in the heat-affected zone.)

Another type of discontinuity found in the fusion zone is shown in Fig. 17. The fissures and voids shown are partially filled with oxides.

Fig. 17 Fissures and Voids in the Added Metal Near the Fusion Zone Partially Filled With Oxides. ×100

Fig. 15 is a view at 25 magnifications at the intersection of fracture, and the outside surface of the howitzer at the origin of rupture. It is to be noted that the rupture did not pass through the weld metal (upper-right-hand zone) but originated in the heat-affected zone just outside the weld metal. This is the region in which the ribbon of hard metal exists, as shown in Fig. 12, and which is prone to cracks and other discontinuities, as shown in Figs. 14, 16, and 17.

It is to be noted that the structure and hardness of the heat-affected zone pertains to metal which has not been stress-relieved. The defects are those expected in welding 0.40 carbon low-alloy steel of heavy cross section without preheating and with undeveloped technique. The welding does not represent current practice.

Proof is now complete that the rupture originated in the heat-affected zone just outside a weld.

The defects of the welded zone furnish the explanation of the failure of the structure. Cracks and other stress raisers were present at the outside surface where the rupture originated. The crack system gradually grew under the stress of firing until complete failure of the structure resulted. The fact that final failure took place on a half pressure round is not at variance with our knowledge of fatigue failures. All the facts related fit together in a single picture and justify the methods used in the analysis of fracture. One last check completes the story. The manufacturing records were consulted and disclosed that the area in question had been welded by metallic arc to correct an error made in machining.

Acknowledgments

In preparing this paper the work of the following members of the Watertown Arsenal Laboratory Staff have been freely drawn upon: M. G. Yatsevitch, P. R. Kosting, H. C. Mann, W. C. Warner, and H. G. Carter, all of whom participated in the investigation of this failure.

The Use of the Piezoelectric Gage in the Measurement of Powder Pressures

By R. H. Kent[1] and A. H. Hodge,[2] Aberdeen Proving Ground, Md.

The development of a piezoelectric gage for the measurement of powder pressures in guns is reviewed. A gage is obtained having a high natural frequency, insensitivity to shock, and accuracy. Details concerning the construction of the gage, the recording apparatus, and the method of calibration are given. Records are reproduced showing pressure waves in a gun and a closed chamber. Such records have been of value in the improvement of the design of powder charges and the determination of interior ballistic data.

No doubt shortly after the invention of cannon in the middle ages, there must have been felt the need of a device for measuring pressures generated in such weapons to enable one to calculate the stresses produced by firings. However, it was not until 1857 that the first practical pressure gage was developed by General Rodman of the Ordnance Department of the U. S. Army. Rodman's gage consisted essentially of a piece of copper which was indented by a knife that was subjected to the pressure in the chamber. The pressure was deduced from the depth of the indentation.

Some time afterward Noble of England invented the well-known crusher-type pressure gage in the form which is now used. In this type of gage, the piston which is subjected to the gas pressure compresses a copper cylinder and the pressure is deduced from the amount of compression. Although such a gage should be capable of producing accurate measurements of pressures which are slowly applied and kept on, serious errors are encountered when it is used to measure the rapidly changing pressures in guns. The deformation of the copper cylinder exceeds the elastic limit and a flow of the copper takes place. It takes a certain time for this flow to be completed and thus in general, the maximum pressures indicated by the copper-crusher gage are too low.

The gages of Rodman and Noble measured only the maximum pressure in the gun. For a complete study of the generation of pressure, a gage is obviously needed which gives the pressure in the gun as a function of the time. For such a recording gage to be a satisfactory instrument for the study of powder pressures, it must satisfy the following requirements:

1 Its natural period must be small compared with the duration of the pressure-time curve. For the 14-in. gun, this duration is about 0.05 sec while for a caliber 0.30 rifle it is about 0.001 sec.

2 The gage must not be affected by the rather violent recoil and vibrations of the gun.

3 Its precision must be of the same order of magnitude as the precision of the velocity measurements.

Vieille made the first recording time-pressure gage about 1884 by the simple expedient of recording on a revolving drum the motion of the piston of the copper-crusher gage as it is compressed. Since that time a number of mechanical and electrical gages have been developed. Of these, the piezoelectric appears to satisfy the listed requirements in a much more satisfactory manner than any other.

The piezoelectric effect was discovered by J. and P. Curie in 1880. They found that certain crystals, e.g., tourmaline, quartz, rock salt, when compressed along certain axes, produce an electrostatic charge proportional to the imposed stress. Obviously such crystals by virtue of their high natural frequency are admirably adapted to the measurement of rapidly varying pressures of short durations such as occur in guns. To record the pressures one simply has to record as a function of the time the charge developed by the crystals. It was not until 1917 however, that the piezoelectric gage was invented by Sir J. J. Thomson[3] and used to measure the pressure in a gasoline engine. Shortly after this his associate Keys[4] used it in measuring the pressure of an explosion. The apparatus of Thomson and Keys consisting of tourmaline crystals and a cathode-ray oscillograph is shown schematically in Fig. 1.

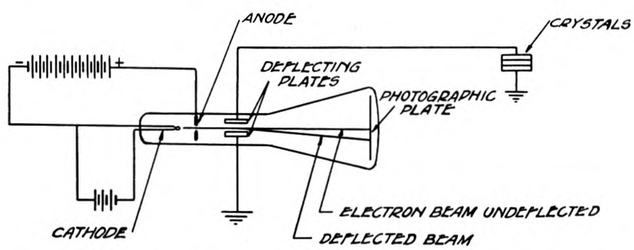

Fig. 1 Apparatus of Thomson and Keys

As the crystals are stressed, the charges produced flow to the deflecting plates establishing a difference of potential between them. The electrostatic field thus arising causes the deflection of the electron beam which is recorded photographically by a plate at the end of the tube.

After the War, the Ordnance Department of the U. S. Army, aware of the experiments of Thomson and Keys, decided to undertake the development of a piezoelectric gage suitable for measuring pressures in cannon.

At that time, to record photographically by the cathode-ray oscillograph it was necessary to place the plate or film within the exhausted cathode-ray tube itself. To obviate this rather laborious technique various other methods were developed in turn for the Department by G. F. Hull, Karcher[5] and Eckhardt, and one of the authors (Kent).[6] Of these the methods of Hull

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Presented at the Annual Meeting of The American Society of Mechanical Engineers, New York, N. Y., December 5–9, 1938.

Discussion of this paper was closed January 10, 1939, and is published herewith directly following the paper.

Released for publication by the Chief of Ordnance, U. S. Army.

[3] "Piezo-Electricity and its Applications," abstract of paper by J. J. Thomson, *Engineering*, vol. 107, April 25, 1919, p. 543.

[4] "A Piezoelectric Method of Measuring Explosion Pressures," by David A. Keys, *Philosophical Magazine*, series 6, vol. 42, October, 1921, p. 473.

[5] "Piezo-Electric Method for the Instantaneous Measurement of High Pressures," by J. C. Karcher, U. S. Bureau of Standards, Scientific Paper No. 445, August 4, 1922, p. 257.

[6] "The Piezo-Electric Gage," by R. H. Kent, *Army Ordnance*, vol. 18, March-April, 1938, p. 281.

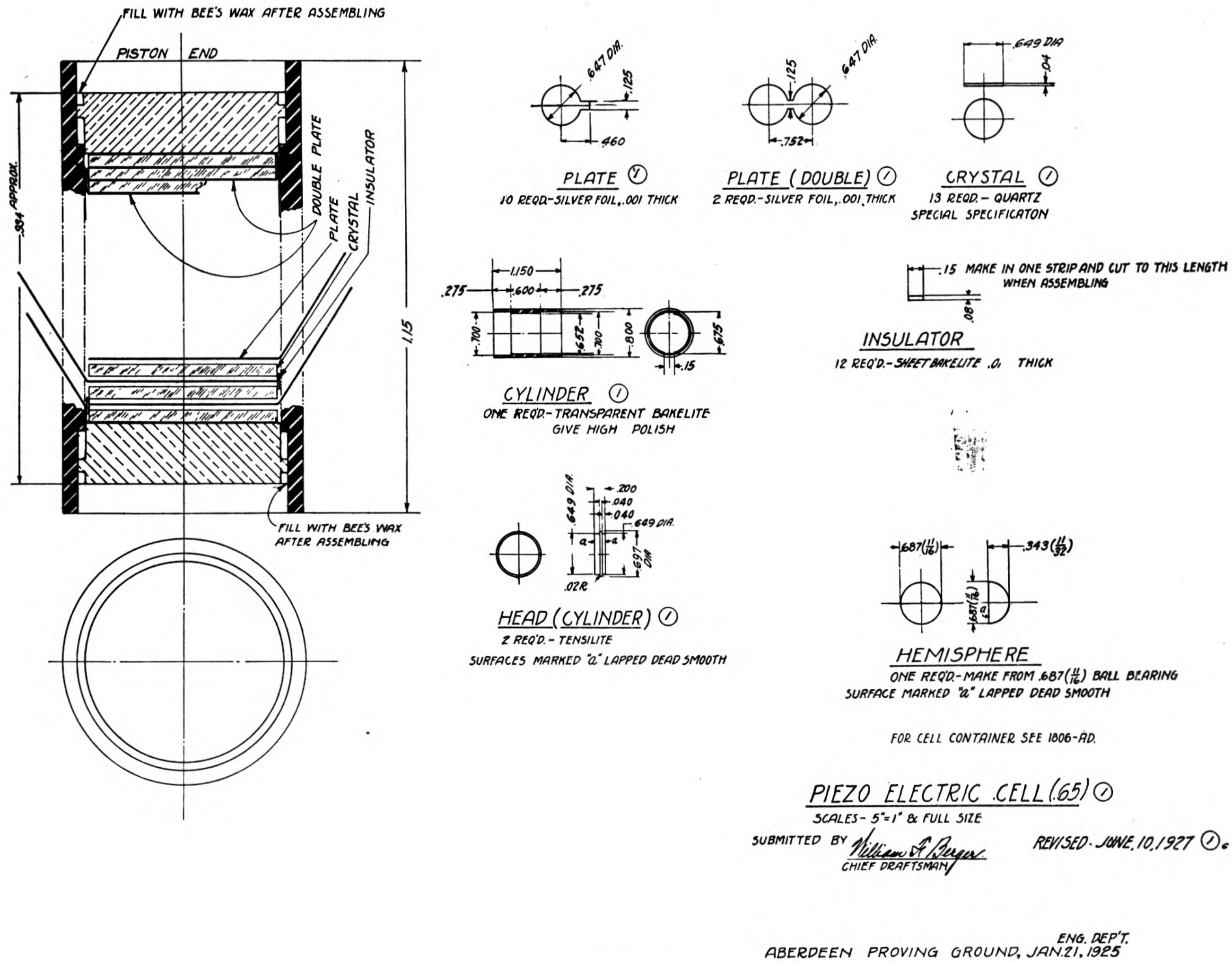

Fig. 4 Piezoelectric Cell

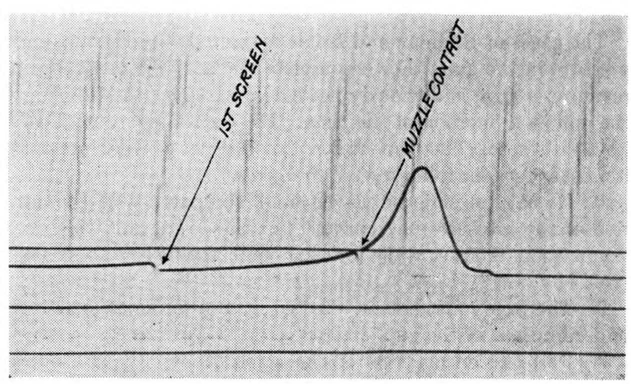

Fig. 3 Pressure Record of Springfield Rifle

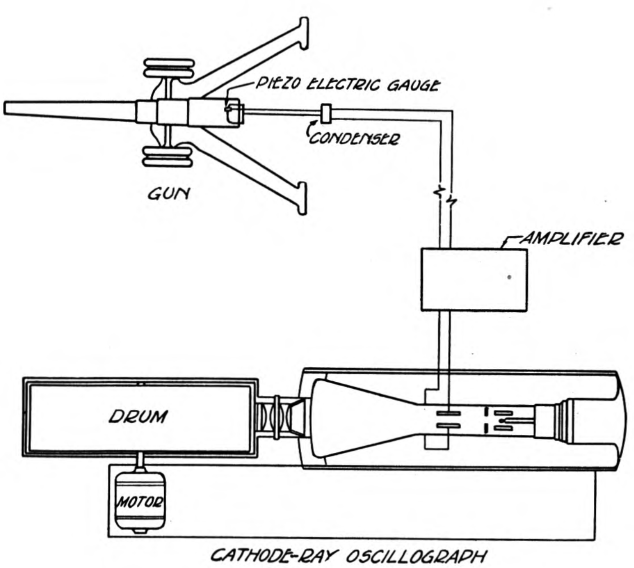

Fig. 2 Diagram of Apparatus

and Kent were essentially similar although differing in important details. The gage was connected to a condenser attached to the input of a direct-current amplifier, the output of which was recorded by a string galvanometer (Hull) or a Duddell electromagnetic oscillograph (Kent).

In 1932, Joachim and Illgen[7] developed for the Zeiss-Ikon

[7] "Gasdruckmessungen mit Piezo-Indikator," by H. Joachim and H. Illgen, *Zeitschrift für das gesamte Schiess- und Sprengstoffwesen*, vol. 27, 1932, pp. 76–79, 121–125.

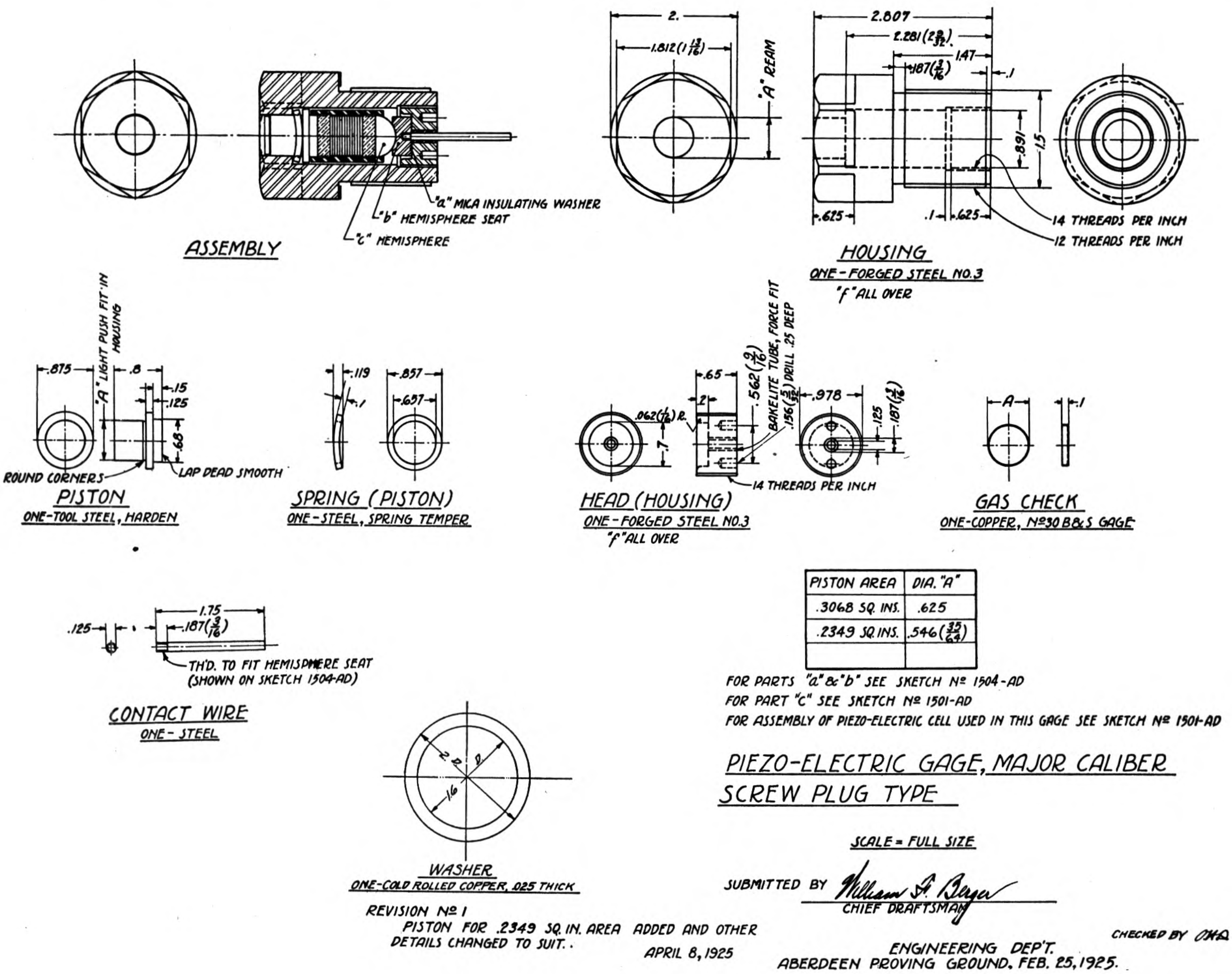

FIG. 5 DRAWING OF GAGE

Company a cathode-ray oscillograph of such characteristics that photographs of the moving spot caused by the impact of the electrons against the fluorescent coating could be taken outside of the tube. This was substituted for the electromagnetic oscillograph referred to. In this apparatus, the amplifier was retained to prevent the loss of charge that would occur with the small capacities required in the absence of amplification.

Since 1934, various other improvements have been made in the gage chiefly by one of the authors (Hodge). A schematic diagram of the apparatus as now most frequently used is given in Fig. 2. The gage consisting of a stack of quartz crystals with its housing is screwed into the chamber of the gun. It is connected to a condenser near by, which is in turn connected by an overhead line or underground cable to the input of a nonreactively coupled amplifier. The output of the amplifier impresses on the deflecting plates of the cathode-ray oscillograph a potential difference which is proportional to the potential difference of the condenser caused by the flow of electrostatic charge from the gage as the pressure changes. As the electrons impinge upon the fluorescent screen at the end of the cathode-ray tube, a luminous spot is produced. By means of a lens an image of this spot is obtained on the film or sensitized paper attached to a revolving drum. Timing lines are also placed on the sensitized paper by means of a tuning fork or other device. A photographic record is thus obtained of the pressure in the gun as it goes off. Fig. 3 is a sample record. It shows the pressure as a function of time in a caliber 0.30 Springfield rifle. The first break in the line to the left shows the time when the bullet reaches the muzzle and the second break shows the time when the bullet arrives at a screen placed in front of the muzzle. The timing lines indicate thousandths of a second. From the record it may be seen that the time from the beginning of the rise of pressure to the exit of the bullet at the muzzle is only slightly more than 0.001 sec.

DESCRIPTION OF VARIOUS PARTS OF THE APPARATUS

The gage proper consists of a piezoelectric cell and a housing in which it is placed. The cell consists of a stack of quartz crystals. The negative sides of the plates are connected together and grounded while the positive sides are connected to an insulated lead-out wire. A drawing of the cell is shown in Fig. 4.

Fig. 5 shows a drawing of the housing with the cell in place. Important features of this are the insulated stem which takes the charge from the positive sides of the plates and the Belleville spring which tends to keep the reaction between the cell and the housing constant during the small displacements of the cell allowed by the elasticity of the housing and the piston when it is subjected to the gas pressure. A photograph of the parts of the gage is shown in Fig. 6.

The gage is attached to a mica condenser of high quality placed near the gun. The capacity of the condenser is varied depending

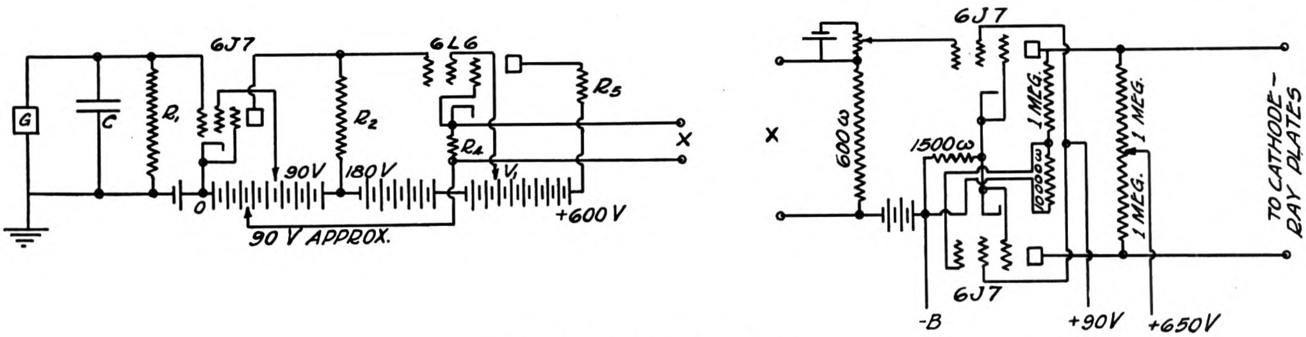

Fig. 8 Amplifiers and Lines

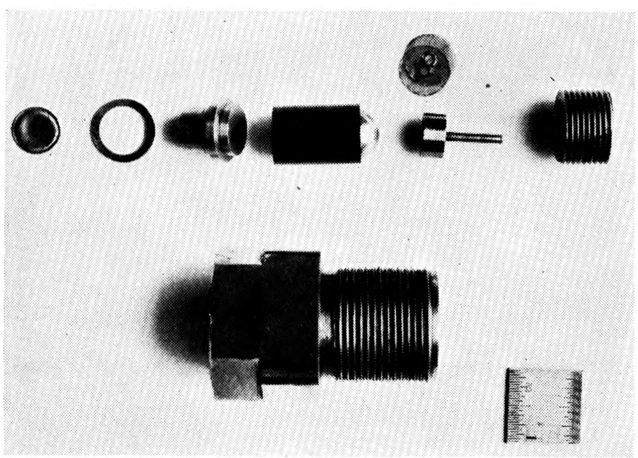

Fig. 6 Photograph of Parts of Gage

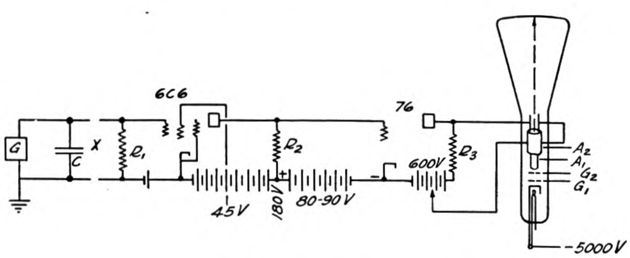

Fig. 7 Diagram of Amplifier

upon the piston area of the gage used and the magnitude of the pressure to be measured. In measuring pressures in ordinary guns, the capacity is between 2 and 3 μf. The condensers are usually connected to an amplifier by means of a line or cable; in some cases a cable of a length of 2000 or 3000 ft is used without any great difficulty for all guns except small arms. In case it is desired to take pressures at more remote points, however, there is installed near the gun, an additional amplifier, which is connected to the amplifier in the laboratory, which in turn is connected to the cathode-ray oscillograph.

A schematic diagram of the amplifier currently used is shown in Fig. 7. In its design an attempt was made to keep the reactances as small as possible, since these cause distortions of the pressure-time curve, and to obtain a uniform amplification of voltages within its range of from 0 to 0.2 volt. No originality is claimed for it. The input tube is of the type 6C6 carefully selected to have a low grid current to minimize the loss of charge.

The batteries supplying the filaments and plate circuits are of such large capacity that the drift of the amplifier is slow. It can be used for a working day without appreciable change in calibration.

The cathode-ray tube is a special one having a blue short-persistence screen. In Fig. 7 the second anode of the tube is shown connected to the amplifier batteries but in case a long line is used between gage and amplifier it is better to use a separate battery for biasing the anode above ground potential.

By means of an f 1.8 lens of 1-in. aperture an image of the luminous spot of the cathode-ray tube is focused on sensitized paper attached to the moving drum.

Timing lines are placed on the moving paper by a tuning fork and an arc light.

The apparatus shown in Fig. 2 with the amplifier shown in Fig. 7 is satisfactory for the measurement of pressures in small arms and closed chambers fired near by in the laboratory and, unless the interference from power circuits and the like is large, for the measurement of the pressures in guns a half mile away. However, for the measurement of the pressures of guns situated several miles from the laboratory, the apparatus of Fig. 7 is inadequate because of the great amount of interference picked up by the long lines. To obviate this it is necessary to amplify the signal before it is transmitted.

Fig. 8 is a schematic diagram of a circuit which has been found to be reasonably good for use with these long lines. This circuit consists of two units, one of which is placed at the gun while the other is in the instrument building. The part of this amplifier which is placed at the gun consists of a high-resistance input tube directly coupled to a tube the cathode current of which passes through a 600-ohm resistor placed across the end of the line. The portion of the amplifier at the camera is a kind of cross between a single-ended and a push-pull circuit. Its input resistance is also about 600 ohms.

Some circuit constants and symbols for Figs. 7 and 8 are:

C = 0.1 to 6 μf
G = gage
R_1 = 5 megohms
R_2 = 0.115 megohm
R_3 = 0.1 megohm
R_4 = 600 ohms, R_5 = 5000 ohms
G_1, G_2 = control grids
A_1, A_2 = electrostatic focusing cylinders
K = cathode
V_1 = 225 volts above cathode connection of 6L6
X = transmission line.

The chief advantage of this circuit is that it provides an output voltage which is relatively large, 400 to 500 volts, and does not tend to cause defocusing of the cathode spot.

The 600-ohm resistors mentioned have this value to prevent reflections at the two ends of the line. Experience has shown that if these resistances differ greatly from 600 ohms, serious distortions caused by reflections will occur in the recorded time-pressure curve. It is a bit difficult to obtain linear amplitude response over the useful range of this amplifier combination, but this defect can be overcome by a careful calibration.

Calibration

To enable the deflection on the oscillogram to be expressed as pressure in psi the apparatus must be calibrated. This is done as follows: (1) The gage itself is calibrated to give the relation between pressure and electrostatic charge, (2) the apparatus, consisting of the condenser, the amplifier, and the cathode-ray oscillograph, is calibrated to give the relation between the charge impressed on the condenser and the deflection of the oscillograph. The combination of the two calibrations gives an over-all calibration of the apparatus. It is possible but impractical to calibrate directly from pressure to deflection. To perform this correctly the dead-weight gage would have to be taken out to the gun. Furthermore, the calibration of the gage remains practically constant but the amplifier has to be calibrated every time records are taken.

The device which is used for calibrating the pressure gage proper is shown in Fig. 9. It is a dead-weight gage of the hydro-

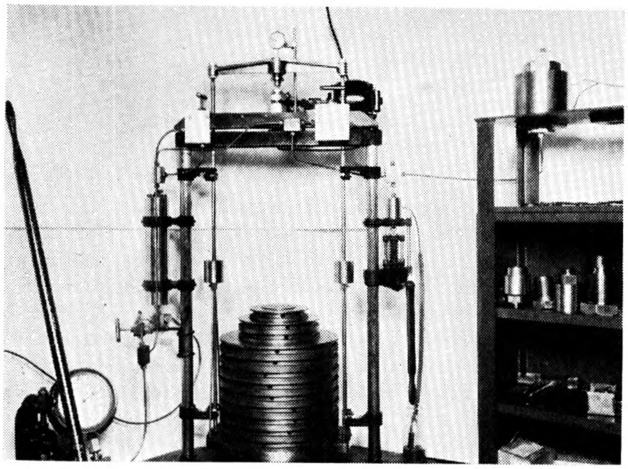

Fig. 9 Dead-Weight Pressure Gage

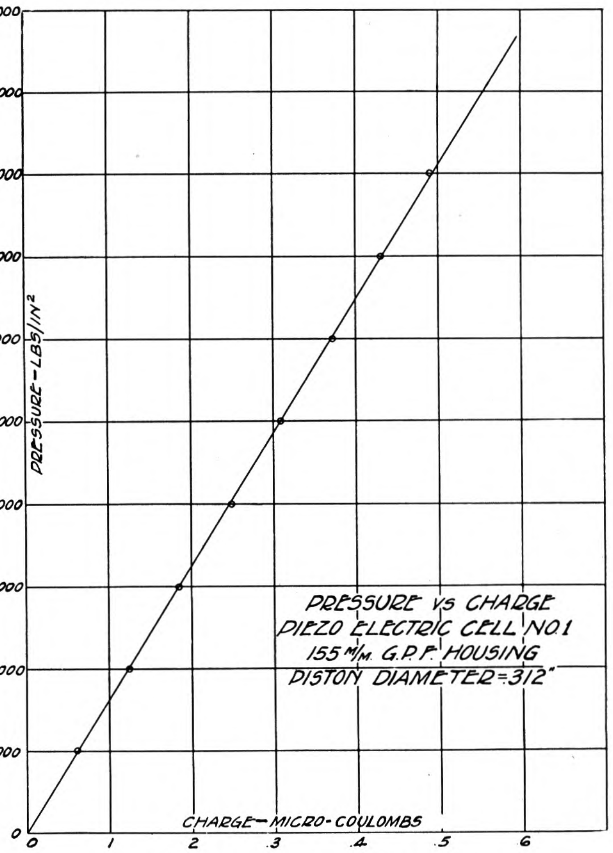

Fig. 11 Pressure-Charge Relation

static type and consists of a hand pump capable of producing 15,000 psi of oil pressure; a pressure intensifier which has a pressure ratio of 10 to 1 between the high- and low-pressure sides and an upper limit of 50,000 psi; an oscillating piston which is caused to rise off its shoulder seat by oil pressure and which raises a pan of from 50 to 500 lb weight; and a quick-release valve which is capable of releasing 50,000 psi pressure and of reducing the pressure against the gage piston to atmospheric in from 0.003

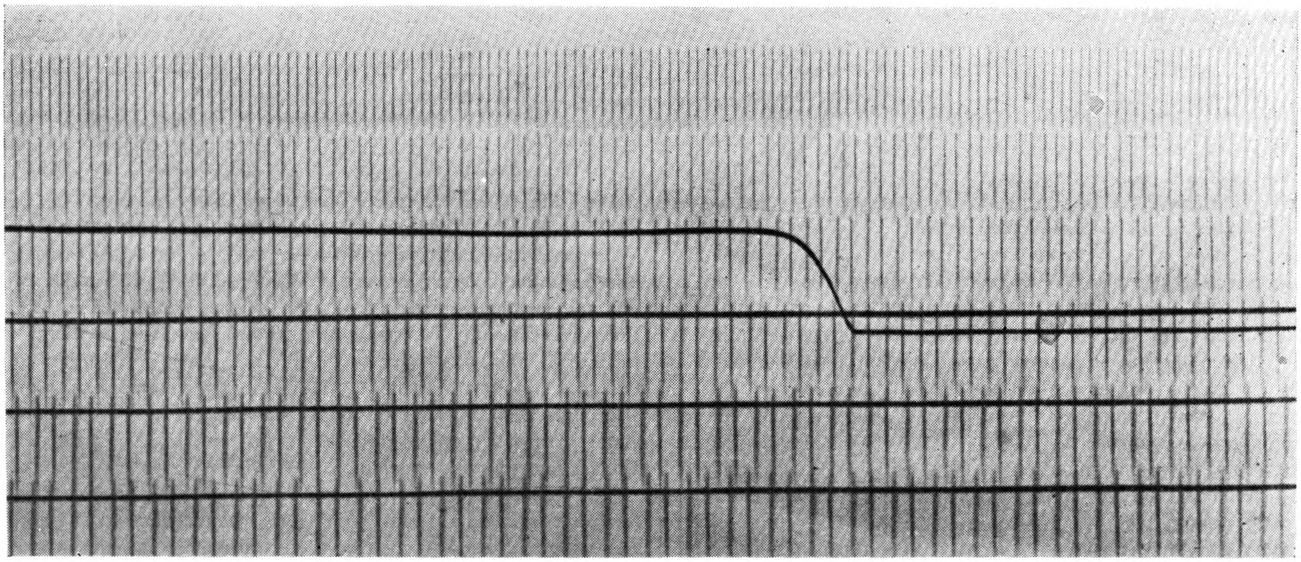

Fig. 10 Curve Showing Release of Pressure

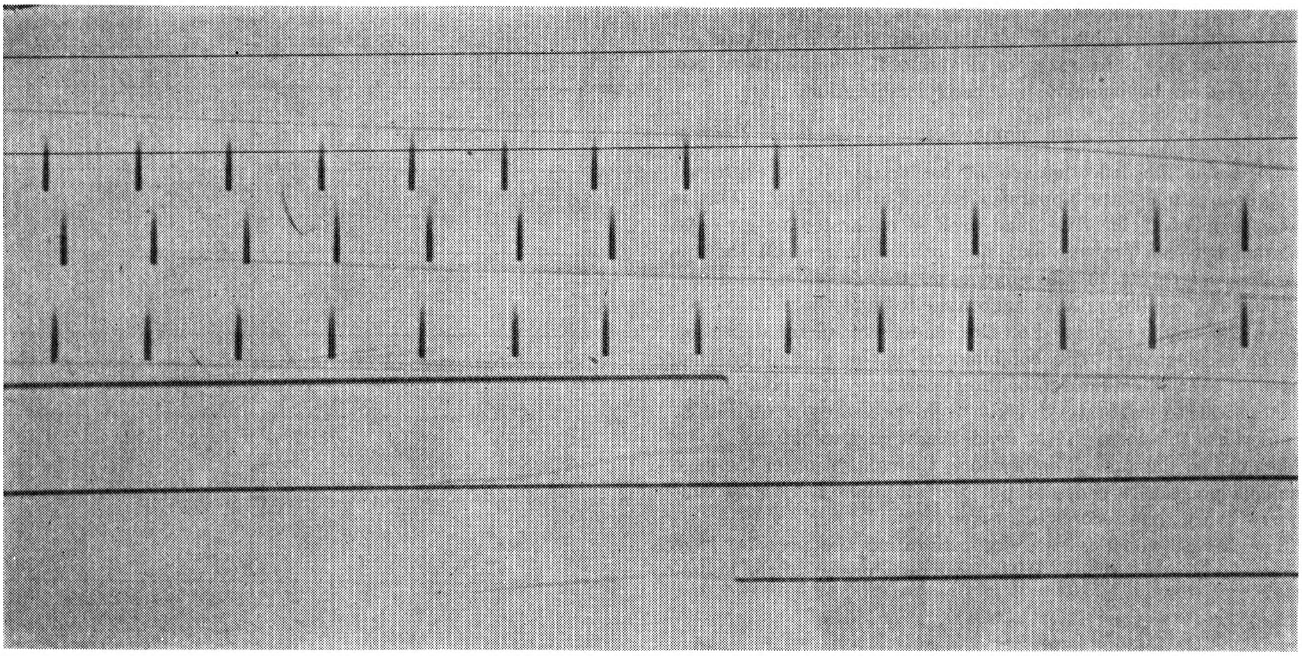

Fig. 12 Calibration Oscillogram

to 0.010 sec. The charge produced by the gage on such a pressure release was used to obtain the pressure-time curve shown in Fig. 10. Thus the charge produced on the quick release of the oil pressure is used to calibrate the piezoelectric gage proper.

Fig. 11 shows the charge produced by the gage as a function of the pressure applied.

Fig. 12 is a calibration oscillogram showing the relation between the deflection recorded on the oscillogram and a charge suddenly impressed on the condensers attached to the input of the amplifier. The fact that the deflection is very sudden indicates that practically no distortion arises from the reactances of the circuits.

The Characteristics of the Gage

As has been previously stated, a gage to be adapted for measurements of pressures in guns should have a high natural frequency, should be insensitive to the shocks produced by recoil, and should have satisfactory precision. The natural frequency of a typical piezoelectric gage is about 50,000 cycles per sec, which is amply high for most guns. If it were desired to do so the frequency could be raised considerably above this by a reduction in the number of quartz plates and in the size of the piston. This would of course reduce the sensitivity of the gage. In so far as sensitivity to the shock of the recoil is concerned, the gage has in general been found satisfactory. No trouble has been found when used in cannon of 3 in. and greater calibers. Some difficulty however has been experienced in a small-arms rifle if the gage is inserted in a weapon which is improperly balanced. The precision while not at the present time as great as desirable is tolerable. The probable error of the measurements is about 1 per cent under ordinary conditions.

In addition to meeting in a fairly satisfactory manner the requirements mentioned, the gage has some other valuable properties which should be mentioned. By reducing the capacity of the condenser, the sensitivity of the apparatus as a whole may be greatly augmented. It is thus possible to make quick shifts from an apparatus designed to measure 50,000 psi to one suitable for measuring 500 psi or less. However, as the capacity becomes smaller, the effects of leakage, referred to later, become more serious, interference increases, and the amplifier may become unstable.

The apparatus may be easily changed so that it will measure directly the rate of change of pressure dp/dt instead of p the pressure itself. For this purpose a resistor is inserted in place of the condenser shown in Fig. 2. With this form of the apparatus all of the charge has to flow through the resistor aside from the negligible amount required to charge the connecting leads. Hence if q is the quantity of charge generated, the current through the resistor is dq/dt and the potential difference between the terminals of the resistor is Rdq/dt if R is the value of the resistance. It is obvious that since q is proportional to the pressure the voltage impressed on the amplifier and hence recorded on the film is proportional to the rate of change of pressure dp/dt. This interesting and valuable use of the gage was developed by Dr. B. S. Mackey of the du Pont Company.

It has previously been mentioned that the input tube of the amplifier is chosen to have a low grid current to minimize the loss of charge. There is of course some loss due to the grid current and also some caused by leakage in the gage, in the condensers, and along the lines. However, when the gage is subjected to pressures of from 25,000 to 60,000 psi, such as occur in guns, it generates a relatively large amount of charge, approximately 0.6 microcoulomb. This large charge permits the use of a condenser of large capacity across the gage terminals. When used in connection with the amplifier of Fig. 7, the capacity is adjusted to keep the maximum input voltage to the line below 0.2 volt, which serves to keep the leakage current low. For pressure cycles which last for more than a few thousandths of a second the charge loss due to the low resistance of the input circuit may become appreciable. However, by the use of an input tube type 6C6 and of mica condensers of good quality it is possible to record pressure-time curves for all ordinary guns with negligible errors due to leakage.

Two of the most serious defects of the apparatus in its present form are: (1) Interference frequently occurs when lines half a mile long or longer are used and (2) the spot of the cathode-ray tube tends to defocus because, with the arrangement shown in Fig. 7, one deflecting plate is kept at constant potential while the

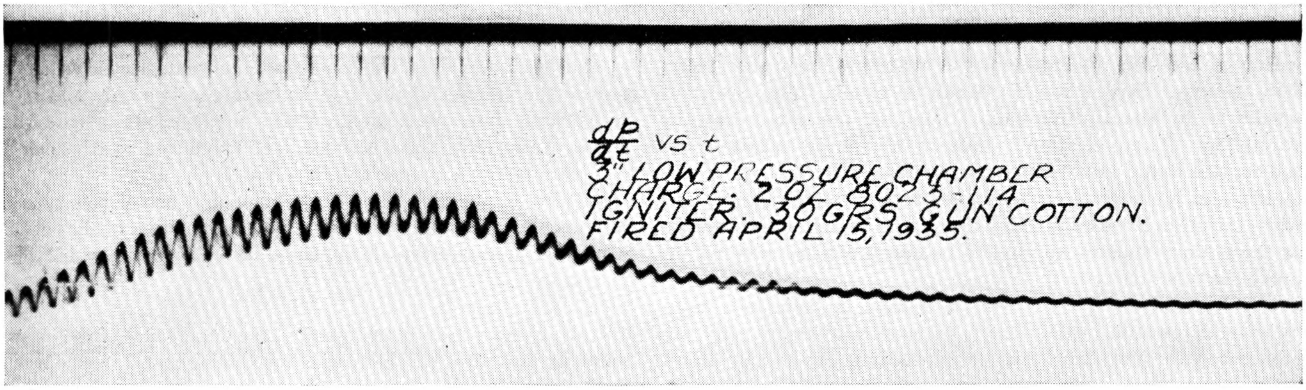

FIG. 14 SHOWING dp/dt FOR A CLOSED CHAMBER

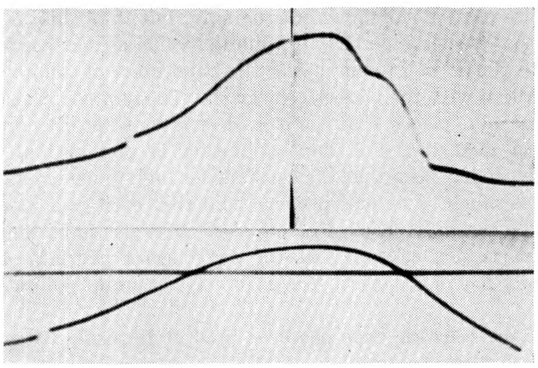

FIG. 13 PRESSURE-TIME CURVES OF 155 MM GUN

potential of the other is varied and this variation causes an appreciable change in the field accelerating the electrons. If push-pull amplification were employed the potential of one plate would be raised and that of the other lowered leaving the potential in the center of the tube unchanged. Push-pull amplification should also reduce the interference on the lines. This interference is partly caused by the unbalanced circuit. If the gage were designed to have both positive and negative terminals insulated a balanced input circuit could be obtained. With this and push-pull amplification most of the effects of interference should be eliminated. Work along these lines is now in progress.

EXPERIMENTAL RESULTS

The piezoelectric gage has been useful to the Ordnance Department in many ways. It has enabled maximum pressures in guns to be measured with much smaller systematic errors than was possible with the copper-crusher gages which had previously been used for this purpose. It has also been of great value in connection with the design of propelling charges for guns. While the possibility that there might be pressure waves of serious amplitude in guns has been known since the time of Vieille, it was not until records were obtained by the piezoelectric gage that it was known that such waves in the chambers of guns were a frequent occurrence, at least in guns using the American type of powder. Although the phenomena are really quite different, the pressure waves in a gun may be compared to the knock in a gasoline engine. If a suitable fuel and design of chamber are used, it is possible to eliminate the knock in the engine. Similarly in a gun, if the chamber, the charge, and the ignition are suitably designed, it is possible to eliminate the pressure waves and obtain a smooth pressure-time curve. Fig. 13 shows two cathode-ray oscillograms of pressures in the 155 mm gun G. P. F. For both of these oscillograms, the amount and kind of powder were the same but the arrangement of the charge was different. As may be seen, with the arrangement used to make the upper oscillogram, pressure waves of considerable amplitude are obtained while the arrangement of charge in the lower oscillogram eliminates the pressure waves completely.

When the apparatus is adapted to measure dp/dt, it is possible to get a sensitivity greatly increased over that obtained when p itself is measured. By the use of the dp/dt apparatus, it is therefore possible to record waves of an amplitude so small that they would probably escape detection if records were made with the ordinary apparatus. Fig. 14 is a copy of an oscillogram showing dp/dt in a closed chamber. From this oscillogram, the frequency of the waves may readily be obtained. From the frequency and the length of chamber the velocity of sound in the powder gas is determined. From the velocity of sound in turn, if the pressure and density are known, the ratio of specific heats γ may be calculated. It has been found that values of γ for the powder gas obtained in this way are in fairly good agreement with those calculated from the specific heats of the constituent gases based upon quantum mechanical theory.

ACKNOWLEDGMENT

In conclusion the authors gratefully acknowledge the assistance and cooperation of W. F. Berger and P. Ewing who designed many of the fittings and gage housings, R. C. Gerdom whose advice and mechanical skill have been invaluable, J. L. Sentman who has built practically all of the amplifiers and who has also taken the many records, and J. S. Durham who has cared for the gages at the guns and assisted with practically every phase of the pressure measurements.

The authors are indebted to the Ordnance Department, U. S. A. for supporting this interesting development and for permission to publish this paper.

Discussion

K. J. DEJUHASZ.[8] This paper is a valuable contribution to the literature on piezoelectric pressure indicators, which instruments have attained a considerable importance in the solution of engineering problems encountered in the measurement of rapidly changing pressures and forces. Though pressure indicators have been built also on other electrical principles (variation of resistance, inductance, capacitance, inverse Wiedemann effect, and photoelectric effect) and the last word has not been said as to which is the best principle, yet it appears at the present stage that piezoelectric indicators best fulfill the many engineering requirements.

[8] Associate Professor of Engineering Research, The Pennsylvania State College, State College, Pa. Mem. A.S.M.E.

At the Engineering Experiment Station of The Pennsylvania State College electrical indicators have been used for the investigation of pressures in Diesel-engine cylinders and in fuel-injection systems. It was found difficult to obtain fully consistent results as to quantitative calibration, due presumably to the variability of amplification. Keeping the amplification under full control would greatly increase the reliability of these instruments. In previous piezoelectric cells one terminal is grounded while the other is led to the amplifier. Insulating both terminals as the authors suggest appears to be one promising way to improve the functioning in this respect.

Development on piezoelectric indicators is being advanced by several research institutions. By some workers, namely, by Meurer[9] at the Dresden Institute of Technology, and by Boerlage, Broeze, Van Dijck and Peletier at the Proefstation, Delft, Holland,[10] it has been found that mounting the quartz crystals between precompressing screws introduces hysteresis effects due to the minute slippage which occurs under stress between the male and female screw threads. Therefore the tendency is to eliminate the screw threads altogether and to mount the crystals between elastically yielding elements. Fig. 15 shows the cell used at Dresden and Fig. 16 that employed at Delft, both of which are built on this principle.

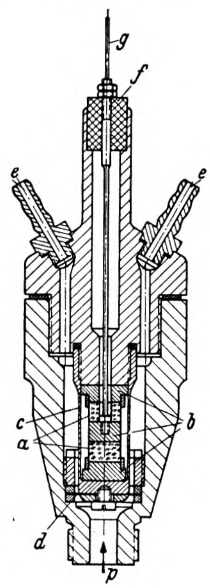

Fig. 15

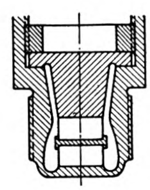

Fig. 16

Fig. 15 Cell Used at Dresden

Fig. 16 Cell Used at Delft

Another tendency in the construction of piezoelectric cells is to reduce the number of the quartz crystals to two only and most recent cells have been built in this manner. It is true that thereby the sensitivity of the cell itself is reduced but this can be compensated by a greater amplification.

In view of the increasing importance and use of this type of instrument, and of the progress in the construction of the cells, it would be of interest to learn the authors' experience in these respects.

[9] "Beitrag zum Bau Piezoelektrischer Indikatoren," by S. Meurer, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 8, no. 5, 1937, pp. 249–260.

[10] "Der Indikator," by K. J. DeJuhasz and J. Geiger, Julius Springer, Berlin, Germany, 1938, pp. 124–126.

B. H. Mackey.[11] In my opinion, the importance of the instrument as described in this paper for research and development work cannot be overemphasized. The present method of measuring pressures with crusher cylinders will probably be used for routine work until the piezoelectric or some other form of pressure gage is simplified considerably, but it is quite obvious that the crusher gage has very limited value as a research tool.

It might be well to emphasize the importance of calibration in connection with the piezoelectric gage. The impression seems to exist in some places, that the piezoelectric constant of quartz is a definite quantity and that calibration is unimportant. Our experience has been that the constant for high-grade crystals can vary as much as several per cent and that each gage assembly must be carefully checked before use. It is also important that the quartz surfaces and those of the collector plates be almost optically flat in order that the gage may retain its calibration in use.

The piezoelectric gage has been used at Burnside Laboratory in essentially the form described in this paper for measuring pressure rises of comparatively long duration, i.e., 1 sec and also for pressure impulses given off by detonating explosives. In the first case, it is necessary to balance out the leakage currents with a separate supply voltage and in the second, it is necessary to damp the gage and the electrical circuit to eliminate extremely high frequency vibrations. The present paper has already indicated the wide pressure range through which this instrument gives satisfactory results.

Authors' Closure

As mentioned in the paper, it is necessary to place a Belleville spring between the head of the piston and the housing to keep the reaction between the housing and the piston approximately constant as the quartz crystals are compressed. This provides the elastically yielding element in the gage as developed at the Aberdeen Proving Ground.

With reference to the number of crystals, it is believed that most of the piezoelectric gages have contained not more than two plates. However, in our work, it is advisable to have many more plates because the gages as a rule are so remote from the amplifier.

[11] Burnside Laboratory, E. I. du Pont de Nemours & Company, Penns Grove, N. J.

The High-Pressure High-Temperature Turbine-Electric Steamship "J. W. Van Dyke"

By LESTER M. GOLDSMITH,[1] PHILADELPHIA, PA.

The author discusses the features of various steamship designs which were considered in the development and final adoption of the construction of the tanker S.S. *J. W. Van Dyke*. After data on steamship designs had been collected and analyzed an intensive study was made of hull form, propeller design, and power-plant requirements. The author describes the results of model tests conducted at model basins in Hamburg, Germany, and Washington, D. C.; the results of these tests were the deciding factor in selecting the propeller and hull designs. The paper includes a complete description of the steam-generating equipment, electrical equipment, high-pressure piping, controls, and miscellaneous electrical equipment which were finally selected for the ship. The author presents graphically the results of trial runs made with this turbine-electric steamship.

FOR a better understanding of the engineering development leading up to the construction of the tanker S.S. *J. W. Van Dyke*, an outline of some of the previous marine-engineering experience of The Atlantic Refining Company will first be given and may serve as a clarifying background for a ship carrying the highest steam pressure and temperature ever to go to sea from the United States.

Twenty years ago this company pioneered in the installation of Alquist or laminated type of double-reduction gears in five ships of 2600 shaft hp each, went through the entire period of this early development in reduction gears, and replaced them finally with compound turbines and single-reduction sets having solid gear wheels and embodying low tooth pressures in the design.

Fifteen years ago the company again pioneered by designing and building the first three Diesel-electric tugs and a 1200-ton Diesel-electric tanker, immediately followed by a 7500-ton tanker with the same system of propulsion. These installations were followed by others until this company had a fleet of fourteen vessels with Diesel-electric machinery, ranging in size from approximately 400 to 3000 shaft hp. At the same time, it was also operating on the same routes a number of the older steam-powered ships having both reciprocating and geared turbine machinery.

In view of this pioneering, and the experience and engineering development which followed it, together with the fact that the owners were manufacturers of petroleum products, it is evident that the company's engineers were absolutely unbiased from an engineering viewpoint, and had no other goal or object than to produce power for propulsion and transportation at the least cost per shaft horsepower, and per ton-mile.

By 1930, experience with, and observation of, Diesel-engine installations had convinced the company that this form of power had definite limitations. In other words, for horsepowers up to 3000, the Diesel engine is an undeniably economical prime mover, but above that size, doubtful, especially in view of the advent of the high-pressure high-temperature steam turbine. Having in mind that the company, within five years, would be faced with the necessity of replacing its then existing tonnage, the author inaugurated a study and thorough investigation of high-pressure and high-temperature steam plants. It was not until 1935, however, that the conclusion was reached that boilers and small turbines, using high-pressure and high-temperature steam, were suitable for marine applications. Keeping in mind the background of experience outlined, studies were made and tabulated of all available types of power plants, including Diesel-electric, direct Diesel, geared-turbine, and turboelectric.

While these data were being collected and analyzed, an intensive study of hull form and propeller design was begun. In order to correlate these studies of the power plant, propeller, and hull a description of the latter is first given.

The problem was to design a modern tank vessel to carry approximately 18,000 tons dead weight on a length of about 520 ft between perpendiculars and a maximum molded draft of 29 ft 6 in. at an average sea speed of $13^1/_4$ knots. This speed was considered the most profitable after a careful study of the conditions under which the ship was to run, and is the point at which a further increase in speed would reduce the margin between operating cost and gross profit.

A hull of the following dimensions was finally settled upon for tank testing and determination of propelling power: Length between perpendiculars = 520 ft; molded breadth = 70 ft; molded draft = 29 ft 6 in.; block coefficient (on between-perpendicular length) = 0.78; and displacement = 23,898 tons.

The Hamburg model experiment tank was chosen for model tests since it is well known to be the largest and best-equipped of its kind in the world and work on the previous ships had been done

[1] Chief Engineer, The Atlantic Refining Company. Mem. A.S.M.E. Mr. Goldsmith was graduated from Drexel Institute with a degree in electrical engineering in 1914, and after serving with the United Gas Improvement Company and the City of Philadelphia, became mechanical engineer with The Atlantic Refining Company in 1916. He was promoted successively to research engineer, engineer of tests, and superintendent of mechanical laboratory. In 1923 he was appointed technical assistant to the president, with the assignment of pioneering the application of Diesel-electric drive to ships. Soon thereafter he became the company's consulting engineer. In 1934, Mr. Goldsmith became manager of the engineering and construction department, and chief engineer in 1937. In addition, he is a director and vice-president of the Atlantic Pipe Line Company, a director of the Atlantic Oil Shipping Company, and chief engineer of the Atlantic Pipe Line Company, Keystone Pipe Line Company, and Buffalo Pipe Line Corporation.

Contributed by the Power Division and presented at the Annual Meeting of The American Society of Mechanical Engineers, held in New York, December 5–9, 1938.

Discussion of this paper was closed January 10, 1939, and is published herewith directly following the paper.

TABLE 1 COMPARISON OF HORSEPOWER REQUIRED FOR COUNTER- AND CRUISER-STERN SHIPS

Speed, knots	Effective horsepower: Counter stern	Effective horsepower: Cruiser stern	Improvement by use of cruiser stern, hp
10	1298	1242	56
11	1742	1676	66
12	2292	2213	79
13	2944	2852	92
14	3756	3625	131

NOTE: By effective horsepower is meant the horsepower determined by towing the model; therefore, the propulsion losses are not included.

TABLE 2 COMPARISON OF TWO PROPELLER DESIGNS

Speed, knots	Effective hp	Propeller No. 1257: Shaft hp	Prop. coef	Speed, rpm	Propeller No. 1275: Shaft hp	Prop. coef	Speed, rpm
10	1242	1853	67.0	70.3	1890	65.6	64.3
11	1676	2490	67.4	78.0	2535	66.1	71.5
12	2213	3289	67.3	85.8	3290	67.2	78.4
13	2852	4294	66.4	93.7	4235	67.3	85.5
14	3625	5410	67.0	101.5	5280	68.6	92.5

NOTE: Horsepowers are metric; multiply by 1.0138 to convert to English units.

TABLE 3 CHARACTERISTICS OF PROPELLERS TESTED AT THE WASHINGTON, D. C., AND HAMBURG, GERMANY, MODEL BASINS

	Washington	Hamburg
Number of blades	4	4
Diameter, ft	19.500	18.170
Effective pitch, ft	15.167	16.540
Disk area, sq ft	298.600	259.500
Projected area, sq ft	118.000	.....
Developed area, sq ft	128.600	116.700
Pitch ratio	0.778	0.910
Developed area ratio	0.431	0.450
Projected area ratio	0.395	.....

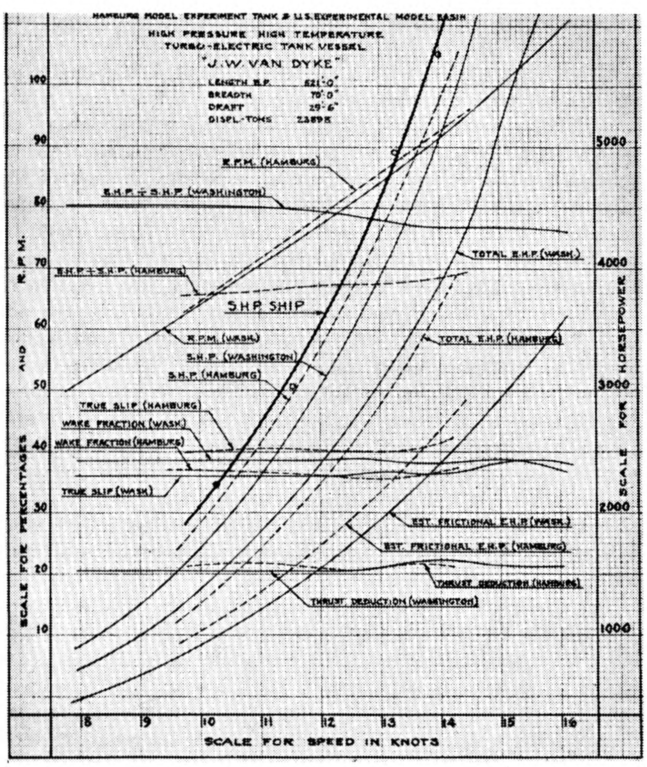

FIG. 1 COMPARISON OF MODEL EXPERIMENTS AND SHIP PERFORMANCE, WASHINGTON, D. C., AND HAMBURG, GERMANY, MODEL BASINS

there. At the time these experiments were started in March, 1934, there was still doubt in the minds of some shipbuilders as to the value of the so-called cruiser stern on the average commercial ship and, in order to obtain conclusive evidence on this point, the model was towed both with cruiser stern and the overhung counterstern, which is now almost obsolete. The results given in Table 1 leave no doubt as to the value of the former in cutting down eddy making and reducing the total resistance by an appreciable amount.

Self-propelled tests were next run with the cruiser-stern model, using propellers designed by the staff at the Hamburg model basin. Two different propellers were tried with results as given in Table 2. It can be seen from Table 2 that propeller No. 1275 gave the best results at the designed speed of 13.25 knots.

As stated previously, these tests were run in March, 1934, and it was not until January, 1936, that the company was ready to go ahead with the construction of the vessel. In view of this lapse of time and the fact that some higher propulsive efficiencies than shown in the Hamburg tests apparently had been obtained in one or two cases in this country, it was decided to have the same model experiments repeated at the U. S. model basin in Washington, D. C. This would serve not only as an interesting check on the results but would also provide opportunity for further possible improvement in efficiency. A model from the same lines was therefore tested at Washington at the same displacement, both for effective horsepower and self-propelled for shaft horsepower with a propeller designed by the staff at the Washington model basin. The results of these tests together with those obtained at Hamburg are shown in Fig. 1, and the principal characteristics of the two propellers are given in Table 3. From an inspection of the curves, it is readily apparent why the propeller developed in the U. S. model basin was adopted.

With regard to the structural plan of the vessel, the chief point of interest is the fact that her entire cargo tank and fuel bunker space, comprising a length of 353 ft, is electrically welded throughout. The bow and stern portions of the ship were riveted in the orthodox manner. The general arrangement otherwise follows more or less usual tanker practice. The ship is a poop, bridge, and forecastle type with one continuous deck and two longitudinal bulkheads and sixteen transverse bulkheads, the cargo space consisting of nine three-compartment tanks with a total capacity of 868,980 cu ft, or 6,500,000 gal. The modern tendency toward streamlining is followed to a moderate extent by rounding the fronts of bridge structure and poop and by giving the stack an ogival section. A general arrangement and profile plan of the ship are shown in Fig. 2, and her general appearance is shown in Fig. 3.

POWER PLANT

After all the analytical work was finally reduced to figures, it was disclosed that it would require 5000 hp to propel this vessel at the desired speed, as indicated by the model-basin work shown in Fig. 1. It was decided that for this horsepower, the form of power promising the lowest over-all operating cost per shaft horsepower was the turbine-electric drive. The reasons for selecting the turbine-electric drive embody more than just propulsion, for it was the request of the management of The Atlantic Refining Company that this vessel be outstanding in its ability to discharge cargo rapidly. In order to meet the requirements for discharging in less than 12 hr, pumps having a rated total capacity of 10,500 gpm at the desired pressure were selected. It was found that approximately 1000 hp would be required for these pumps when discharging in port. In addition, the trim of the vessel necessitated location of this cargo-pump room amidships. Therefore, it can be seen how simple it is to obtain this power from a standard turbine generator and operate the pumps electrically. It must also be kept in mind that not only economy of operation, but first cost and reduction of weight should be in the mind of every designer of any transportation equipment.

To meet existing rules of the American Bureau of Shipping, the Bureau of Marine Inspection and Navigation, and the require-

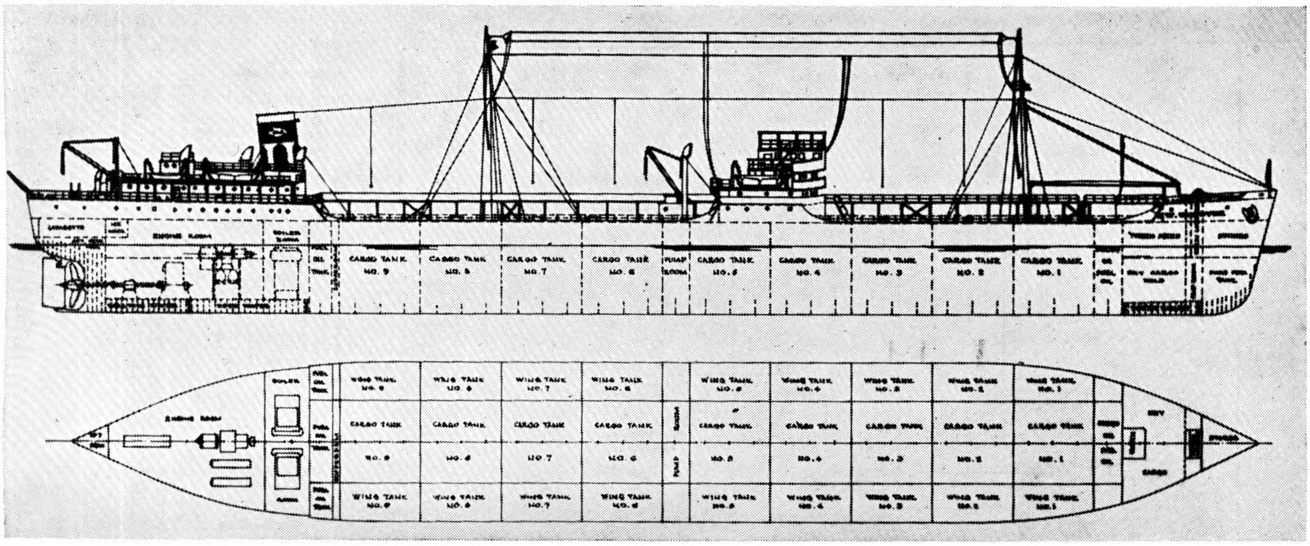

Fig. 2 General Arrangement and Profile of the S.S. *J. W. Van Dyke*

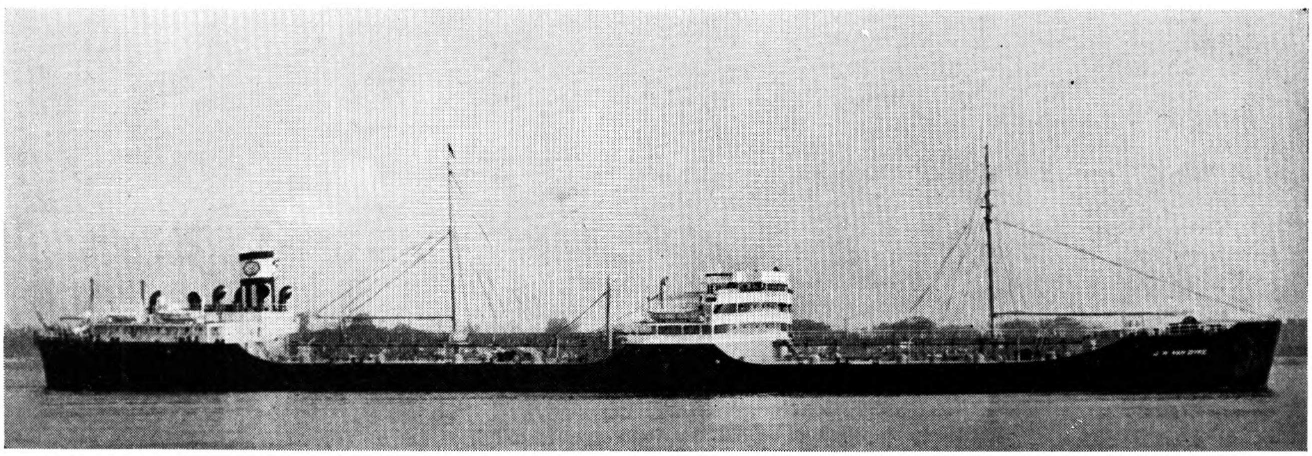

Fig. 3 The S.S. *J. W. Van Dyke*

ments of economy in general, it was decided that the turbine should be designed for a pressure of 600 lb per sq in. and a steam temperature of 825 F. When the decision was made to go to this high pressure and temperature, one problem eliminated by the use of motor instead of gear drive was that of the reversing turbine and the question of keeping it cool.

In the tabulation of the engineering design of the previous turbine-electrically propelled vessels, both in this country and abroad, it was disclosed that none utilized a generator of standard voltage and frequency as we know them to be standard, thinking of land power plants in this country, nor did they employ the use of alternating current for auxiliary uses as is the case in all land plants. As no reason could be found why other than standard equipment should be used, engineering work was started with the idea of profiting by the principles in use in every modern land power station. A complete heat balance, as shown in Fig. 4, was first worked out in connection with the machinery manufacturers.

Three bleed stages were decided upon as giving the best compromise between the ideal and the practical complications of space requirements and piping. Another consideration borne in mind was that evaporation of feedwater should be reduced to a minimum and that every possible precaution must be taken to protect the boilers from impure water; therefore, all clean drains are collected and returned through a cooler into the main condenser while drains which are subject to possible contamination are reevaporated. The evaporators are planned to float on the line and normally only one of them, operating at much below its rated capacity, will be required. Under sea conditions, the heat of the evaporator vapor is used to supply such low-pressure steam as may be required for general heating purposes throughout the ship, and the remainder goes to the intermediate feedwater heater so that this heat is recovered. An emergency vapor line to the condensers is also provided for port use; but with proper operation this should not normally be necessary.

All three boiler feed pumps are turbine driven. Only one is required for normal operation. The exhaust is added to the second-stage bleed.

Therefore, it will be noted that under normal conditions the heat from the evaporator vapor and the feed-pump exhaust is either recovered in the feed or otherwise usefully employed.

No heat which can be put to use is rejected to the sea water except under unusual conditions.

Table 4 gives a comparison of the heat distribution as planned and as actually obtained. It will be noted that an over-all heat efficiency from burners to shaft of 22.5 per cent was expected.

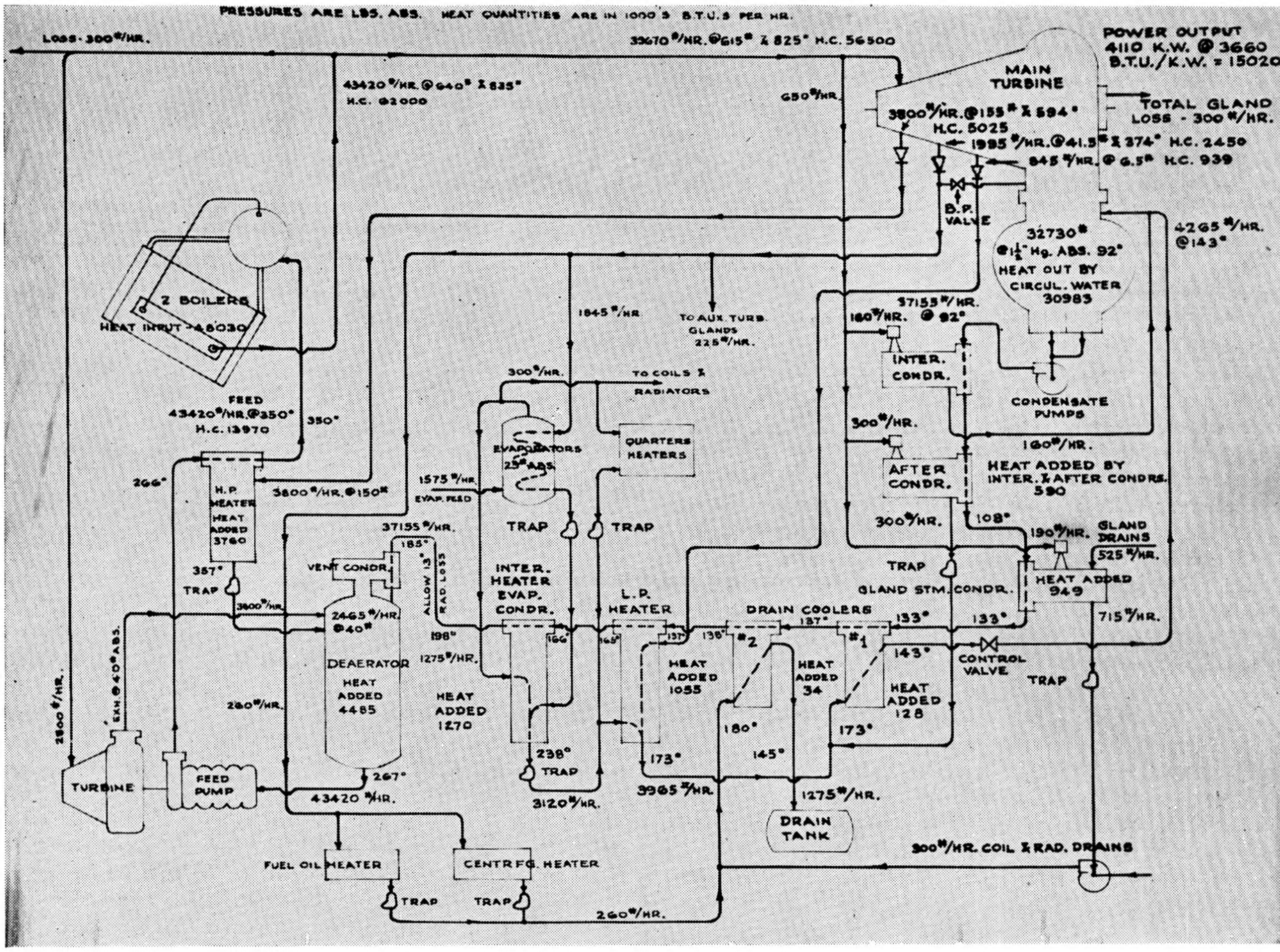

FIG. 4 HEAT-BALANCE DIAGRAM

TABLE 4 SUMMARY OF MATERIAL AND HEAT BALANCE COMPARISONS

LOCATION	DESIGN CONDITIONS		#1 FUEL RUN - 4 HRS.		#2 FUEL RUN - 3 HRS.		J.W. VANDYKE - 4 DAYS VOYAGE #12 - 8/28 TO 9/1		J.W. VANDYKE - BEST DAY VOYAGE #12 - 8/29 TO 8/30		ROBERT. H. COLLEY 8 HOUR TEST RUN		DERIVATION OF QUANTITIES NOTE - ALL INSTRUMENTS WERE CAREFULLY CALLIBRATED FOR THESE TESTS.
	QUANTITY	HEAT 1000 BTU/HR.	QUANTITY	HEAT	QUANTITY	HEAT	QUANTITY	HEAT	QUANTITY	HEAT	QUANTITY	HEAT	
FEED WATER TO BOILERS	43420	13970	43625	14108	44023	14130	45700	14950	45300	14780	45920	14980	BY METERS
HEAT INPUT BY BOILERS	-	48030	-	48715	-	48960	-	50150	"	49800	"	50920	FROM INSTRUMENTS
TOTAL FROM BOILERS	"	62000	"	62823	"	63090	"	65100	"	64580	"	65800	-
TO FEED PUMP TURBINES	2800	3996	2950	4260	2970	4260	2980	4250	2980	4250	3000	4890	CALCULATED FROM PUMP & TURBINE CURVES.
AIR JET VAC. PUMPS	650	928	700	1010	700	1004	700	998	700	998	700	1002	FROM MANUFACTURER'S FIGURES
AUXILIARY SET	RUN AS	MOTOR FROM	MAIN	GENERATOR									
UNACCOUNTED FOR LOSS	300	576	375	353	173	136	470	602	320	442	120	558	BY DIFFERENCE
TO MAIN TURBINE	39670	56500	39600	57200	40180	57700	41550	59250	41300	58900	39300	56900	BY METER
FROM - #1 BLEED	3800	5025	3530	4750	3720	4980	4300	5770	4170	5600	3560	4810	CALCULATED FROM TEMPERATURE RISE IN H.P. HEATER.
#2 BLEED	1995	2450	1820	2200	1844	2250	1222	1490	1390	1697	1835	2240	CALCULATED FROM TEMP. RISE IN INTER. HTR. & DEAERATOR.
#3 BLEED	845	939	0	0	0	0	0	0	0	0	0		
GLAND LOSSES ETC.	300	336	300	336	300	336	300	336	300	336	300	336	MANUFACTURER'S FIGURE
TO CONDENSER	32730	32730	33950	35614	34316	35934	35728	36834	35440	36447	34000	35054	BY DIFFERENCE
TO POWER		15020		14280		14200		14820		14820		14460	FROM MFGR'S FIGURE OF 3660 B.T.U. (93.4 % MECH EFF.) PER K.W.HR.
SHAFT HORSEPOWER	5000		4980		4940		5080		5080		NO METER FITTED		BY TORSION DYNAMOMETER
LBS. TOTAL STEAM/S.H.P.HR.	8.68		8.76		8.95		8.99		8.92				
% HEAT FROM FUEL TO SHAFT	22.5		22.6		22.4		22.7		22.7				
LBS. TOTAL FUEL/S.H.P.HR.	.596		.594		.598		.592		.591				NOTE - THIS IS TOTAL FUEL FOR ALL PURPOSES AND AUXILIARIES. HEAT VALUE OF FUEL = 19000 B.T.U./LB.
% FUEL FOR AUXILIARIES	14.4		15.5		1.54		13.7		13.3				
LBS. FUEL/S.H.P. HR. (FOR PROPULSION ONLY)	.510		.502		.508		.511		.512				

During the trials, we obtained efficiencies as high as 22.6 per cent and in actual operation have obtained 22.7 per cent. It should be noted that these figures are that portion of the total heat supplied by the fuel to the burners which is used for propulsion, and not the gross steam-cycle efficiencies, which figures are 26.4, 26.1, and 26 per cent, respectively. Auxiliary sets do not enter into the normal heat-balance picture, since under usual load conditions they are electrically driven from the main generator.

Steam Generation

The design of the steam-generation plant is as follows:

Boilers are of the single-pass straight-tube header type as shown in Fig. 5, embodying four decks of superheat, stud-tube side walls, and air preheater.

Header-type boilers with air heaters, but without economizers, were selected with the thought in mind that the extreme simplicity of this design would be reflected in maintenance cost and, in particular, that with this type of boiler the tubes can be plugged off or even renewed without loss of ship time waiting for the boiler to cool down enough so that the drums can be entered.

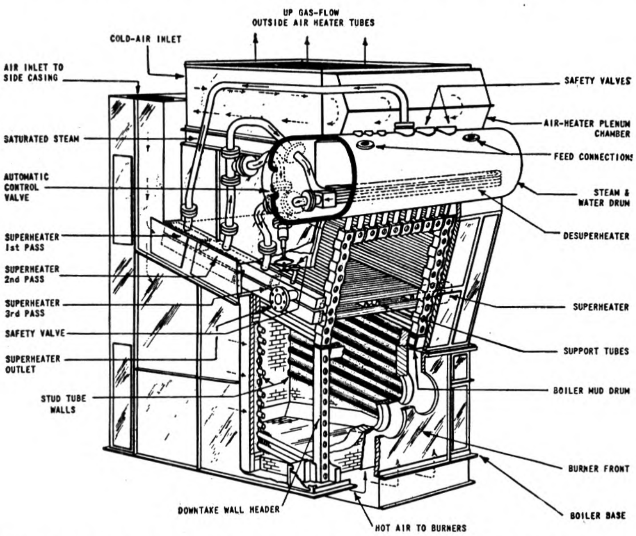

Fig. 5 Single-Pass Straight-Tube Header-Type Boiler Used on S.S. *J. W. Van Dyke*

It will be noted that the arrangement is further simplified by the fact that the boilers are so designed that the gases rise straight through, no baffles being required, thus further reducing internal maintenance.

The boilers have a rated capacity of 22,500 lb per hr each, at a pressure of 625 lb per sq in. gage and a steam temperature of 835 F, with feedwater at 300 F. They each have a total water-heating surface of 3222 sq ft, a superheating surface of 1094 sq ft, and an air-heating surface of 2412 sq ft. The furnace volume is 426 cu ft per boiler, giving a liberation rate of 66,000 Btu per hr per cu ft.

The lower two rows of tubes in both halves of the superheater are of 9 per cent chromium alloy and the boilers can be safely operated at 900 F final temperature.

They are fitted with automatic air-puff soot blowers.

The boilers are provided with complete automatic combustion control, including automatic superheat control, and the charts as shown in Figs. 6 and 7, will show how well this equipment functions. The charts in Fig. 6 are from one of the trial runs and show a continuous record of a 3-hr fuel test at constant load followed by three sets of runs at approximately full load, but under different ballast conditions. Note that no attempt was made to protect the boilers by gradual slowing or speeding up of the main turbine, but in spite of this, the superheat temperature falls off instead of rising when the load is dropped and is held at the set temperature when load is applied. At low loads the boilers are on hand operation; hence, the spread between the air-flow and steam-flow records.

When the low-load condition is to be maintained for any length of time, the burners can be changed to suit and the system again put on automatic operation.

The arrangement of the superheaters and their control is interesting. The superheater is divided into two sections, and a desuperheating coil is located in the drum which is so baffled that the entering feed passes along this coil. The saturated steam from the drum passes through the first half of the superheater and then is divided by a three-way valve so that a variable portion goes through the desuperheating coil before passing on to the second half of the superheater. The three-way valve is controlled automatically by the temperature at the superheater outlet.

It is interesting to note from precedent in this respect that today a land installation of a steam-generating plant, not embodying some form of automatic combustion control, would never be considered. Yet, in land plants, technical men, usually in the operating force and always in the supervisory force, are immediately available, while in the marine field, the supervision is only obtained at long distance. The successful operation of this type of equipment, as demonstrated on the S.S. *J. W. Van Dyke* will undoubtedly mean much greater application because of this demonstration. How well this control functions under sea conditions is shown by the charts in Fig. 7.

High-Pressure Piping

In the design of the high-pressure steam piping, the rules of the Bureau of Marine Inspection, which, at that time, not only prohibited welding, but were written without provisions for temperatures above 750 F, were somewhat of a handicap. However, through the cooperation of the bureau, rulings were obtained which permitted using stress-relieved welded joints. The main steam connections to the boilers, the boiler header, and the line to the main turbine are of 0.5 Mo, 0.2 C seamless-steel tubing with an inside diameter of 5 in. and an outside diameter of $6^1/_2$ in. This is heavier than necessary on account of the Bureau rules for thickness which did not permit taking full advantage of the greater strength of molybdenum tubing at high temperatures. In order to reduce radiation loss as well as the number of joints, the equipment was arranged to reduce the high-pressure steam piping to a minimum. The total length of the main steam lines, including expansion bends and valves, is as follows: Length of boiler risers, the 5-in. header, and main-turbine connection = 72 ft; length of the 3-in. Butterworth heater connection = 21 ft; length of the $2^1/_2$-in. boiler-feed-pump main = 22 ft; and the length of the 2-in. auxiliary-generator connections = 32 ft. All the high-pressure steam piping is of carbon-molybdenum steel. All flanges are recessed or tongue-and-grooved to give confined gaskets. All bolts, including those in the feed discharge lines, are of high-tensile class-C steel.

Main Turbine Generator

The main turbine generator is rated at 4500 kw, 60 cycles, 2300 v at 3600 rpm when supplied with steam at the throttle at 600 lb per sq in. gage and 825 F total temperature with the exhaust at 28.5 in. Hg vacuum.

The normal output for 5000 shaft hp was expected to be 4200 kw, 3850 kw going to the motor and 350 kw to the auxiliaries.

The turbine is of the impulse type, having 15 stages with bleeds after the third, seventh, and twelfth stages. The rotor is a solid forging with the wheels integral with the shaft, the first wheel carrying a double row of blades and all of the remaining wheels carrying single rows. The case consists of a high-pressure alloy fabricated steel section and a cast-iron low-pressure section, the division occurring between the tenth and eleventh stages. The case is also split horizontally to permit removal of the rotor. The blading of both wheels and diaphragms is of chrome-iron alloy; the first-stage nozzles are of steel with chrome-iron alloy partitions and the diaphragms are of cast iron or steel, depending on

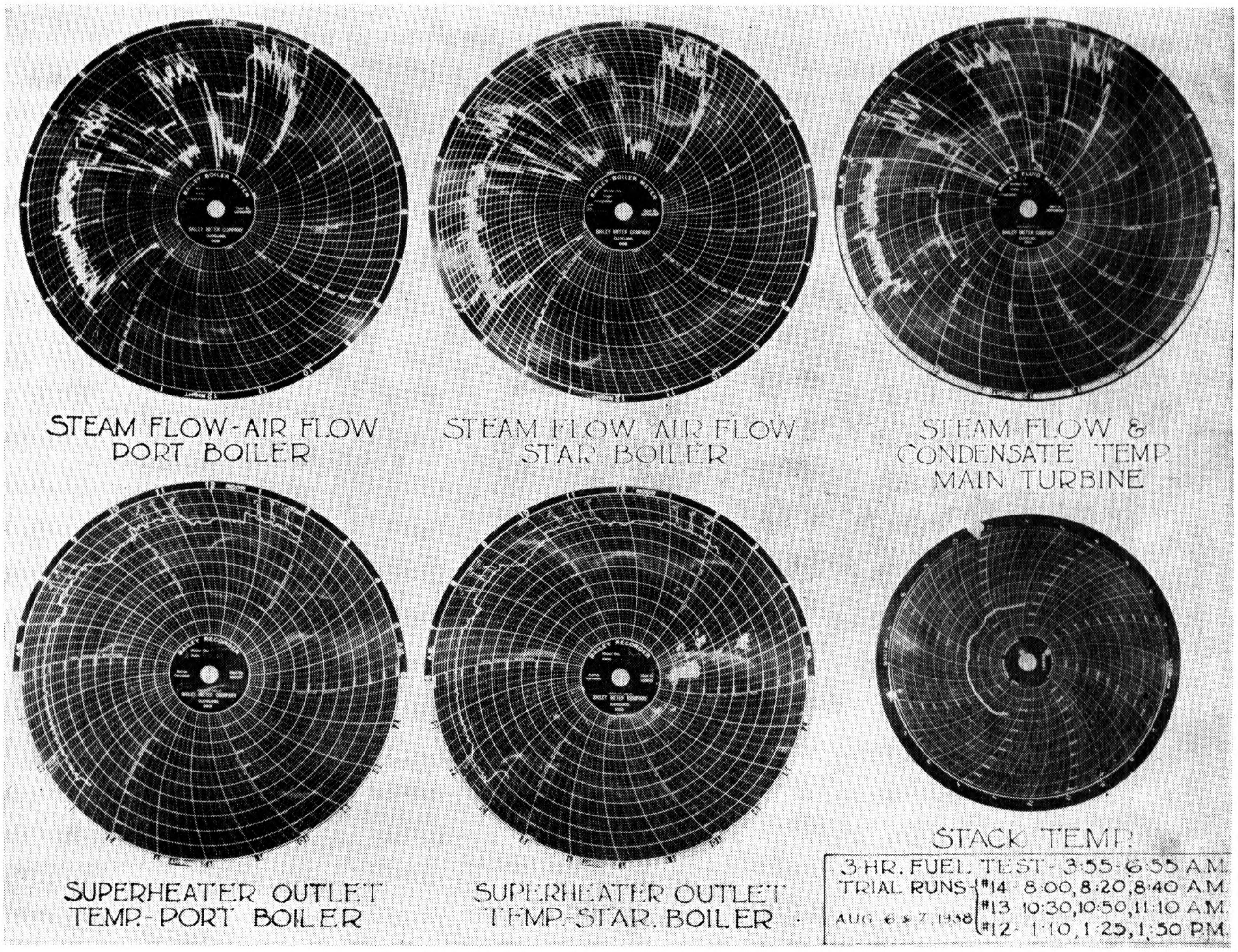

Fig. 6 Chart Record of 3-Hr Fuel-Test Run

the temperature. The packings are of the labyrinth type, and the glands are steam-sealed and provided with an automatic steam regulator.

Governing is accomplished by a valve chest containing six valves leading to the nozzle plate and so arranged that the nozzles are successively cut in or out as the turbine demand increases or falls off, thus avoiding the loss which would be caused by throttling the supply. The speed range which is obtainable is from 900 to 3780 rpm corresponding to 22.5 to 94.5 rpm of the propeller.

The generator, which is directly connected to the turbine, is of the rotating-field type. The stator consists of six coils Y-connected and brought out to six terminals. The rotor is ventilated by fans integrally mounted, which force air into its ends and out at the center; it then passes down through the air cooler, which is in turn water-cooled. This cooler is mounted directly under the generator and is housed in the same case. The bearings are of the plain babbitted type.

The combined turbogenerator set is only 22 ft 9 in. long × 10 ft 3 in. wide.

Propulsion Motor

The propulsion motor is of the synchronous-induction type with 80 poles and is rated at 5000 shaft hp at 90 rpm, 2300 v, 3 phase, unity power factor.

It is only 17 ft diameter and 14 ft long over-all. This length includes the space allowed for sliding the stator aft to clear the rotor in case repairs are necessary. This arrangement avoids any need of disturbing the bearings or shaft alignment; this is the first time such an arrangement has been used.

The motor is separately cooled, but motor, fan, and air cooler are all enclosed in a tight duct system so as to give practically the effect of a totally enclosed motor.

Both the generator and motor are equipped with CO_2 connections for fire protection.

Control

Figs. 8 and 9 show the main control panel with the operating levers. The method of control is extremely simple. Referring to Fig. 8, the left-hand lever (30) as you face the controls has three positions for both ahead and astern movement. As it is moved from the off position, it first closes the line contacts to the generator, then the generator field contacts so that the motor starts as an induction motor, and finally the motor field contacts, which causes the motor to pull into step and run as a synchronous motor.

Beside this lever is the turbine-governor lever (31) which permits varying the speed of the generator from 900 to 3780 rpm by adjustment of the governor setting.

In addition, there is an emergency throttle lever (32) which permits operating the generator on hand control up to 75 per cent of normal speed and an emergency trip (27).

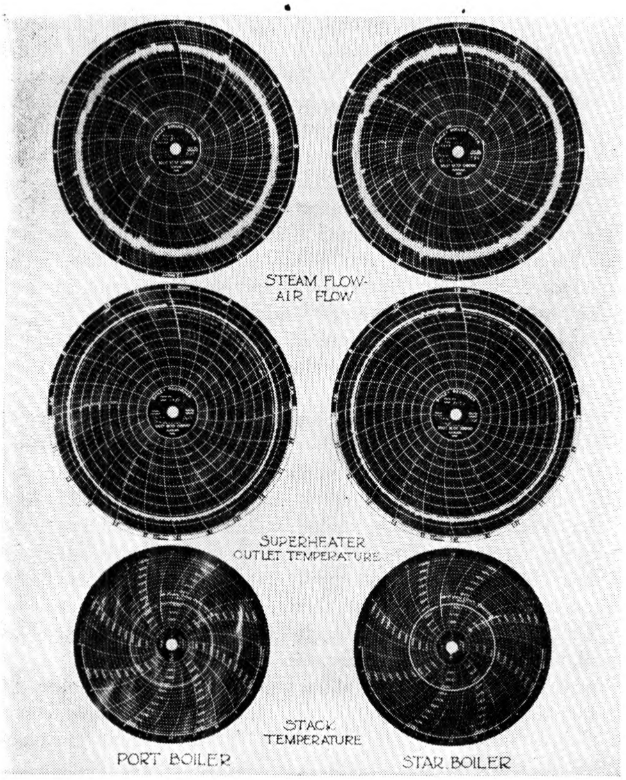

Fig. 7 Chart Record of Boiler Operation Under Normal Sea Conditions

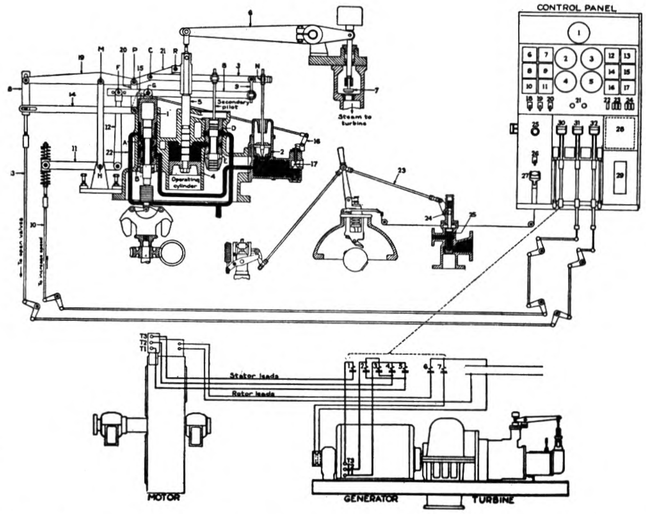

Fig. 8 Schematic Diagram of the Propulsion Equipment

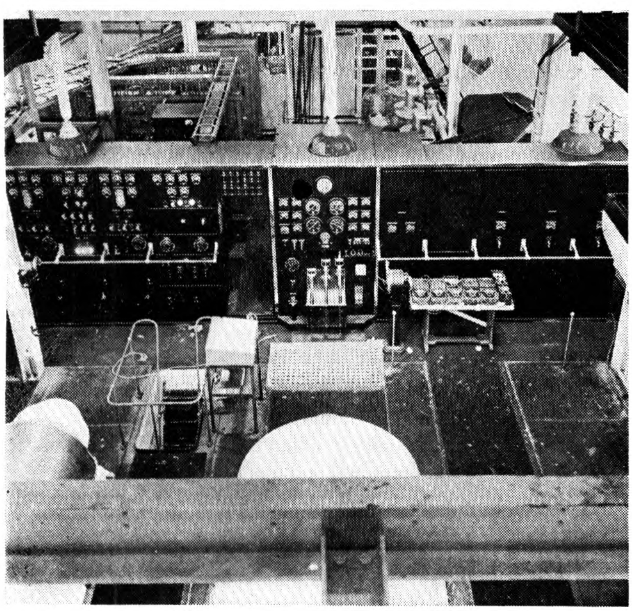

Fig. 9 Control Panels of the S.S. *J. W. Van Dyke*

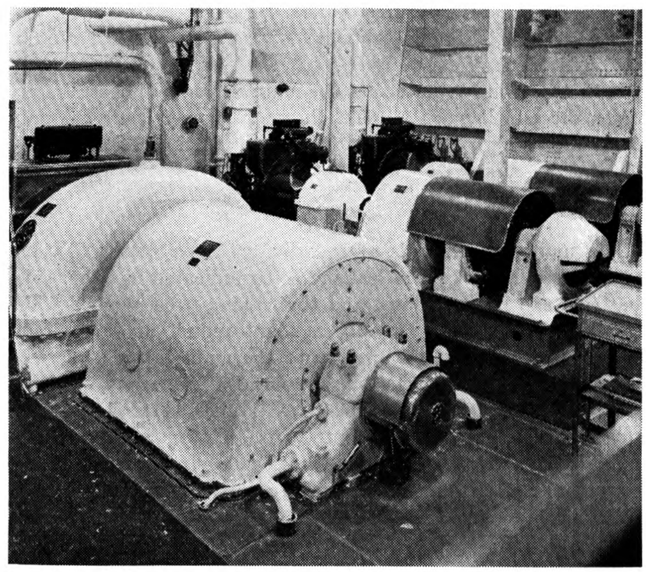

Fig. 10 The Main Generator and Two Auxiliary Sets

All these levers are mechanically interlocked to prevent incorrect operation.

Auxiliary Sets

There are two auxiliary sets, one to run and one for a stand-by unit. Each consists of a 5568/1200-rpm geared turbine driving a 350-kw 60/50-cycle 450-v synchronous generator, a 65-kw 110-v d-c exciter for the main units and a 20-kw 120-v d-c generator for excitation of the 350-kw auxiliary generator and supplying miscellaneous direct-current requirements.

The auxiliary units are so arranged that under normal conditions (60/50 cycles) the 350-kw unit operates as a motor, taking current from the main generator to drive the direct-current machines and drag the turbine. (No cooling of the turbine is required as vacuum is maintained through a connection to the main condenser.) When the speed of the main generator drops to approximately 52 cycles, the speed governor of the auxiliary set automatically starts to open the steam-admission valve so that the auxiliary turbine assumes the full auxiliary load by the time the main unit has slowed to 50 cycles. The circuit breaker between the two generators opens automatically to prevent the auxiliary generator attempting to motor the main generator. When the auxiliary turbine is to operate for any length of time, it is then brought up by hand to 60-cycle speed. When the main unit is again above 52 cycles, the auxiliary generator can be returned to motor operation as follows: The main and auxiliary generators are synchronized by means of a synchronizing device on the auxiliary-turbine governor. The circuit breaker is then closed and the speed governor reset at its lower limit (52 cycles) which cuts off the steam, but leaves it ready to open and causes the turbine to reassume the load automatically when the main generator speed again drops below 52 cycles.

The compactness of both the main and auxiliary units and the convenience and "shipshape" arrangement of machinery made possible by electrification are clearly shown in Figs. 9 and 10.

Fig. 10 is a view from above the switchboard looking forward and to starboard across the main generator and the two auxiliary sets.

Fig. 9 is a view from the bridge across the forward end of the engine room and looks down and aft. In this view the main generator is in the center foreground with the auxiliaries to the left and the steam-gage panel board to the right. Across the center is the switchboard with the main control panel in the center, the three cargo pump panels at the right, and the auxiliary panels at the left. At the extreme left beyond the auxiliary panels may be seen the control panel for all the various motors throughout the ship. In this connection, note that equipment cannot be started from this panel, but only by means of push-button stations located at the equipment. Behind the switchboard on the left may be seen the relays and on the right the 2300-v–440-v oil-filled transformers. The table full of instruments in front of the switchboard is a temporary installation of test instruments.

It is interesting to note that the main motor is almost directly under the panel board so that the main power leads are only about 15 ft long from generator to switch panel and down to the motor.

Condensers

The main condenser is of the divided type with a divided hot well so that in case of leakage being indicated in either side, that side can be shut down and the leaking tube plugged while operation continues with only a slight reduction of vacuum. This condenser is bolted directly to the main turbine, most of its weight being taken by spring supports.

It has 4775 sq ft of $^3/_4$-in. 70–30 Cu-Ni tubes, 14 ft long between tube sheets, solidly rolled at both ends. Expansion is provided for by bellows joints in each side of the shell. It is rated at 35,300 lb per hr at a vacuum of 28.44 in. Hg with sea water at 76.8 F which is the average for the conditions under which this ship is to operate.

The auxiliary condenser has only a single bundle of 800 sq ft effective surface. The tubes are of the same material, size, and length as the main unit so that only one set of spares is required. As in the main unit, the tubes are rolled in at both ends and expansion is provided for by a bellows in the shell.

This unit has a capacity of 6000 lb per hr at a vacuum of 28.44 in. Hg with sea water at 76.8 F. This corresponds to the output of one auxiliary unit at 105 per cent load.

It can also handle the output of both units at 70 per cent load (8000 lb per hr) at a vacuum of 28.1 in. Hg.

The auxiliary condenser is connected by a header to the main condenser, and the auxiliary turbines exhaust into this header. Valves are provided to shut off either auxiliary unit and to permit the operating unit to exhaust either into the auxiliary or the main condenser.

The arrangement of the circulating-water system is unique and merits some explanation.

It has long been recognized that the principal cause of condenser-tube corrosion is entrained air. Many attempts have been made to eliminate the trouble by such means as water-box vents, streamlining of water passages, and flaring of tube ends, all of which have helped, but have not solved the problem.

Therefore, an attempt has been made to arrange the sea suction so that the entrained air is removed before the circulating water reaches the pumps. In order to accomplish this, a chamber has been provided between the tank top and the skin of the ship. The sea connections are made into this chamber. Baffles and vents are provided so that any entrained air is separated and removed before the water reaches the intakes of the circulating pumps, which are at the bottom of this chamber.

Examination of the condenser tubes in the S.S. *J. W. Van Dyke* after 6 months of operation indicates that this is proving effective as there was no sign of any pitting of the tubes. The impellers of the circulators were also examined and likewise showed no signs of corrosion or erosion.

It is believed that this arrangement will prove to be an important contribution to the protection of both condensers and circulating pumps.

The circulating pumps are of the so-called "deep-well" type, having mixed-flow type impellers at the lower end of vertical shafts driven by hollow-shaft vertical motors. Each pump normally supplies one side of the condenser, but they are manifolded so that both sides of the condenser can be supplied from one pump (when this is done, there is a loss of vacuum of about $^1/_2$ in.). These pumps also supply the generator and motor air coolers and the lubricating-oil cooler.

The remaining engine-room machinery is not described in detail, but a summary list is given in Table 5.

For those who may fear that the use of turboelectric equipment instead of gear drive adds unduly to the machinery weight, it will be of interest to note that the main turbine generator weighs 91,000 lb; the main motor weighs 107,000 lb; the air coolers and motor-cooling fan weigh 6100 lb; and the propulsion control equipment weighs 7600 lb. The total weight of this equipment is 211,700 lb.

The total dry machinery weight of the entire ship is only 423 tons, including deck machinery and cargo pumps.

The total weight of the power plant and auxiliaries only is 405 tons (wet). This equals 0.08 tons per shaft hp. The usual figure for a geared-turbine ship of this power is from 0.11 to 0.12 tons per shaft hp (550 to 600 tons); for direct Diesel, or Diesel-electric drive, it is from 0.11 to 0.14 tons per shaft hp (550–700 tons).

This low machinery weight is reflected in the unusually high ratio of dead weight to displacement obtained for the ship, the figure being 0.758, which is considered very good, especially in view of the extra-large cargo-pumping equipment installed.

Expressed in another way, the lower machinery weight alone permits the carrying of from 1000 to 2000 bbl more cargo.

Cargo Pumps

As stated earlier, it was necessary to provide cargo-pumping equipment of sufficient capacity so that the ship could be unloaded between tides. This required pumps of a total capacity of 10,500 gpm, giving an actual pumping time of approximately $10^1/_2$ hr.

In order to take advantage of the ship's generating equipment, it was decided to use deep-well pumps equipped with 2300-v motors as with this arrangement the motor room can be completely separated from the pump room except for the shafts which pass through stuffing boxes at each end of the cover pipe with bleedoffs to the suction side between them.

The pumps are three-stage centrifugals rated at 3500 gpm at 1750 rpm versus pressures of 80 lb per sq in. on gasoline and 90 lb per sq in. on crude oil, and requiring 220 and 245 hp, respectively. At maximum load on salt water, the power requirement is 300 hp. No stripping pumps are used, but instead, the cargo pumps are set in suction boxes which in turn are connected to vacuum priming pumps through float boxes so that as long as a

TABLE 5 SUMMARY OF MACHINERY USED ON THE S.S. "J. W. VAN DYKE"

Number of units	Unit	Capacity or size	Power
2	Boilers	22500 lb per hr at 625 lb per sq in. and 835 F	....
..	Boiler controls	....	Air operated
..	Soot blowers	....	Air operated
1	Main generator	4530 kva, 2300 v, 0.994 power factor 60 cycle, 3 phase	....
1	Propulsion motor	5000 shaft hp at 90 rpm, 2300 v, 60 cycle, 3 phase	....
2	Generator and motor air coolers	2300 sq ft (each)	....
1	Ventilating fan for main motor	....	15 hp
1	Main condenser:		
	Capacity	35300 lb at 28.44 in. Hg vacuum	....
	Area	4775 sq ft	....
2	Auxiliary generator sets	430 kw, 450 v, 3 phase, 60 cycle, 0.8 power factor	....
1	Auxiliary condenser:		
	Capacity	6000 lb at 28.44 in. vacuum	....
	Area	800 sq ft	....
4	Low-pressure heaters and drain coolers	40000 lb feed to 167 F	....
1	Deaerator	43500 lb feed to 267 F	....
1	High-pressure heater	50000 lb feed to 350 F	....
2	Evaporators	1400 lb per hr, each	....
2	Fuel-oil heaters	3000 lb per hr, each, to 235 F	....
1	Lubricating-oil cooler	40 gpm; 160 to 130 F	....
1	Butterworth heater and drain cooler	200000 lb per hr of water 50 to 180 per cent with 150 lb steam	....
1	Centrifuge oil heater	75 gal per hr; 60 to 140 F	....
1	Evap feed grease extractor	3000 lb per hr, fresh water	....
2	Main circulating pumps	3750 gpm each	30 hp, 1150 rpm
1	Auxiliary circulating pump	1200 gpm	10 hp, 1150 rpm
2	Main condensate pumps	90 gpm	15 hp, 3500 rpm
1	Auxiliary condensate pump	20 gpm	5 hp, 3500 rpm
3	Main feed pumps	125 gpm at 755 lb per sq in.	125 hp, 3600 rpm
3	Main-feed-pump turbines	125 hp, 3600 rpm	....
2	Evaporator feed pumps	1500 lb per hr each	1 hp, 1750 rpm
2	Fresh-water pumps	1000 gal per hr	$1^1/_2$ hp, 1750 rpm
2	Fire and Butterworth pumps	425 gpm vs. 100 lb per sq in., each	....
2	Engine-room and forward bilge pumps	325 gpm	$7^1/_2$ hp, 3500 rpm
1	Sanitary pump	90 gpm	5 hp, 3500 rpm
2	Fuel-oil service pumps	8 gpm vs. 300 lb per sq in. each	$7^1/_2$ hp, 1150 rpm
2	Fuel-oil transfer pumps	150 gpm vs. 100 lb per sq in. each	15 hp, 1750 rpm
1	Lubricating-oil pump	59 gpm vs. 50 lb per sq in.	3 hp, 1160 rpm
1	Heating-system vacuum pump	4 gpm and 3 cfm of air	1 hp, 1750 rpm
3	Main cargo pumps	3500 gpm gasoline vs. 80 lb per sq in.	300 hp at 2300 v (1750 rpm)
1	Pump-room bilge pump	160 gpm vs. 50 ft head	3 hp at 1750 rpm
3	Cargo priming pumps	90 cfm each at 18 in. vacuum	$7^1/_2$ hp at 1150 rpm
2	Air compressors	60 cfm each vs. 120 lb per sq in.	15 hp at 870 rpm
3	Forced-draft fans	8500 cfm each vs. 4.5 in. draft	10 hp at 1750 rpm
2	Boiler-room ventilating fans	11500 cfm each vs.0.1 in. draft	$1^1/_2$ hp at 1750 rpm
4	Engine-room ventilating fans	14650 cfm each vs. 0.15 in. draft	2 hp at 1750 rpm
1	Thrust bearing (roller type)	116000 lb thrust ($16^1/_8$ in.)	....
1	Lubricating oil centrifuge	75 gal per hr	$^1/_2$ hp, 1750 rpm
1	Desuperheater (for Butterworth)	22000 lb of steam per hr	....
1	Meter for steam to turbine	50000 lb per hr, max	....
1	Gland steam condenser	725 lb of steam + 210 lb of air per hr	....
1	Refrigerating machine	$2^1/_2$ ton	$7^1/_2$ hp, 1750 rpm
3	Capstans	....	25 hp, 865 rpm
2	Deck winches	....	30 hp, 200 rpm
1	Electrohydraulic windlass	....	75 hp, 1775 rpm
1	Electrohydraulic steering gear	....	Two 20 hp, 875 rpm
1	Two-unit gyro pilot system	....	....
1	Gyrocompass	....	....
1	Salinity recorder	4 point 0–10 grains	....
2	Butterworth sets	....	....
All	Auxiliary motors	....	440 v, 3 phase, 60 cycle

suction box is filled with liquid, the suction to the vacuum pump is closed by the float. If air enters, it passes into the float chamber, the valve opens and the air is drawn off through the vacuum pump, thus restoring suction to the cargo pump. The intakes of the cargo pumps being close to the bottoms of the suction boxes, the pumps continue to operate at full capacity until the tanks are practically empty as is evidenced by the fact that the *Van Dyke* has discharged her cargo in as short a time as 10 hr and 6 min, and normally discharges in less than 11 hr. Her sister ship, the S.S. *Robert H. Colley*, established a record of 9 hr and 21 min to discharge her cargo of 138,069 bbl of crude oil.

Miscellaneous Electrical Equipment

Motors. All pump and fan motors are of the class-2 marine type and are totally enclosed except (*a*) the main motor which is ventilated by a separate motor-driven fan and (*b*) motors in locations classified as hazardous, which are of the class-1 type and are explosion proof. This latter group includes the cargo pump, the cargo priming pump, and pump-room bilge-pump motors. The deck-machinery motors are also of class-1 explosion-proof construction.

All motors except the steering-gear motors are provided with overload relays.

The steering-gear motors, and the lights and galley service are

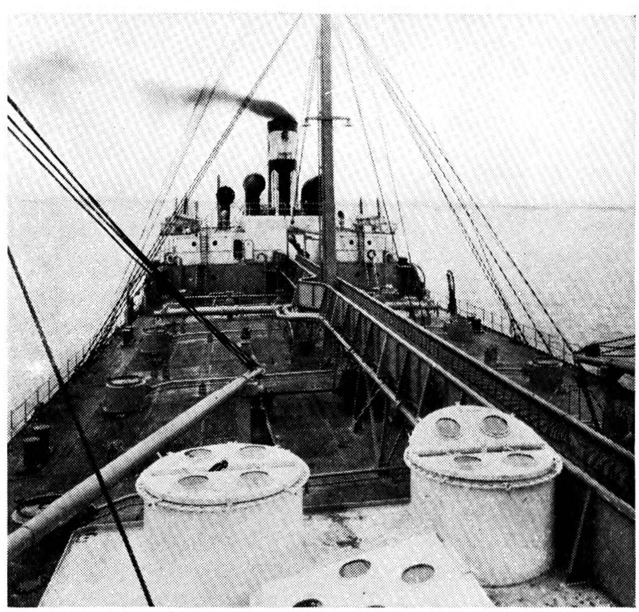

FIG. 11 Fore-and-Aft Gangway Has Solid Sides

on an automatic transfer between the two auxiliary units. Certain other motors are equipped with sequential relays to restart automatically after a power failure in the following order: (1) main lubrication oil pumps, (2) main circulators, (3) forced-draft fans, (4) main condensate, and (5) fuel-oil service.

In connection with the electrical equipment, the problem of carrying the cables forward to the cargo pump room and the midship house was given considerable thought, and it was finally decided to provide the fore-and-aft gangway on deck with solid sides instead of open railings and to run the cables along the inside of these where they are not only amply protected but are easily accessible. This gangway, with its cables, is shown in Fig. 11. The 2300-v cables are further protected by a box cover which can barely be seen where the cables turn down to enter the cargo-pump motor room in the foreground.

Steering Gear

The steering gear proper is of the conventional electric-hydraulic type with horizontal opposed rams connected to a tiller on the rudder stock and with power furnished by duplicate Hele-Shaw type oil pumps driven by 20-hp, 875-rpm induction motors. The steering control, however, is somewhat unusual in that there is no mechanical shaft or hydraulic telemotor connection between the bridge and steering-gear compartment, either for normal or emergency steering. The system used consists of two entirely independent electrical controls using Selsyn motors, and separate circuits from bridge to steering gear. One of these controls is arranged for either hand or automatic steering through the usual gyropilot and is a full follow-up self-synchronizing system. The alternate control is a nonfollow-up type operated by means of a small lever on the right-hand side of the steering stand. Any one of these three steering methods, automatic, hand, or emergency nonfollow-up is instantly available to the helmsman without moving from his position by simply shifting the three-point selection lever on the left side of the steering stand. At the afterend of the ship, for operating the steering-gear hydraulic valve, are

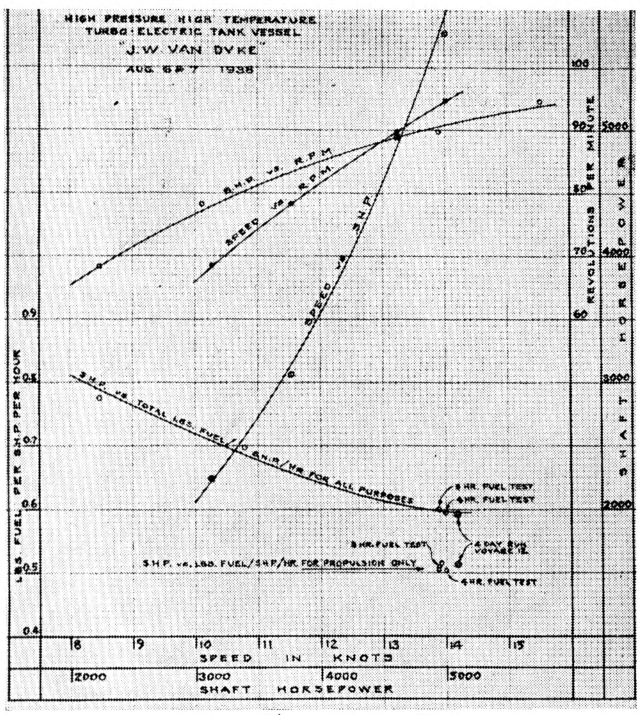

Fig. 12 Performance Curves From Tests of the S.S. *J. W. Van Dyke*

two Sperry power units, one connected to the automatic and normal hand-steering control just described, and the other to the alternate nonfollow-up control. Each of these power units is connected on its output side to the hydraulic valve mechanism through a magnetic clutch operated by the selector lever on the bridge steering stand so that the unit not in use is disconnected.

In addition to the described steering controls from the bridge, the ship also has two emergency steering stations aft, one on the aft end of the poop deck house and the other on the poop deck close to the stern. The former is a mechanical rotating shaft control directly to the hydraulic valve on the steering gear and the latter is the usual hand gear operating directly on the rudder stock through spur and worm gearing and a quadrant.

Trial Results

General. On August 6 and 7, 1938, after having been in service about 5 months, the ship was put through a series of speed, horsepower, and fuel-consumption tests over a measured mile at the Delaware breakwater and off the Delaware capes. The results of the horsepower and speed tests are shown by the heavy curve in Fig. 1 and the performance curves in Fig. 12. Fig. 1 also shows the model-basin results. It can be seen that the ship-performance curve comes within the usual allowance of 12 to 15 per cent added to effective horsepower as obtained on the model, to allow for such effects as sea conditions. It should also be noted in this connection that the ship was fitted with bilge keels while the model was tested without them and the depth of water over the trial course was not as great as is generally considered desirable if speed is not to be affected. The empirical formula

$$\frac{10 \times \text{draft in ft} \times \text{speed in knots}}{\sqrt{(\text{ship length in ft})}}$$

gives a close approximation to the minimum desirable depth and, in the present case, the actual depth was about 60 to 70 per cent of the 180 ft given by this formula.

Minimum turning-radius tests were made both to port and to starboard at full speed and showed a diameter of turning circle of 0.47 nautical mile to port and 0.45 mile to starboard; the time to turn through 180 deg was 3.6 min in each case.

A test of the steering gear showed that, with the ship at full speed, in full-loaded condition, the helm could be put from hard-over to hard-over in 17.8 sec with a maximum load on one of the two 20-hp steering-gear motors of not much more than half its rated capacity.

With the ship in full-loaded condition and going full speed ahead, a full-speed-astern bell stopped her headway in 5.08 min in a distance of 0.40 nautical mile.

A similar test, in which the propulsion motor was stopped instead of being reversed, gave results in which the ship drifted for approximately 1 mile in 14.05 min before stopping.

Both of the last two tests were made against a moderate wind with a velocity of approximately 13 knots per hour and against a tidal current of possibly 1 knot per hr.

Power Plant. It was of course impractical to arrange for the measurement of every quantity and temperature throughout the entire steam and water cycles as indicated on the heat-balance diagram shown in Fig. 4, but boiler meters were provided, as was a recording and integrating flowmeter for the steam to the main turbine. Thermometers and pressure gages were installed at all important points so that it has been possible to determine fairly closely and without any unwarranted assumptions how close the actual results came to theoretical expectations.

Boilers. It was possible to determine the output and efficiency of the boilers quite closely. Fig. 13 shows the expected performance from which it will be noted that the boilers were designed to give an efficiency of 85.5 per cent at the rated load of 22,500 lb per hr actual output at a pressure of 625 lb per sq in. and a temperature of 835 F; this corresponds to an equivalent evaporation of 26,260 lb per hr. During the trials, two fuel-test runs were made, one of 4 hr and one of 3 hr duration. During the first of these runs, the boilers were operated at a pressure of 620 lb per sq in. and a temperature of 855 F, producing 21,810 lb per hr each, or an equivalent evaporation of 25,100 lb per hr. The feed temperature was 351 F. The stack temperature was 324 F, CO_2 was 14.7 per cent, CO was 0, and oxygen 1.8 per cent. These results gave an efficiency of 86.6 per cent, a stack loss of 11.3 per cent and an unaccounted-for loss of 2.1 per cent.

The second run was at almost exactly the same rate and conditions, and showed an efficiency of 86.9 per cent with stack losses of 11.3 per cent and unaccounted-for losses of 1.8 per cent.

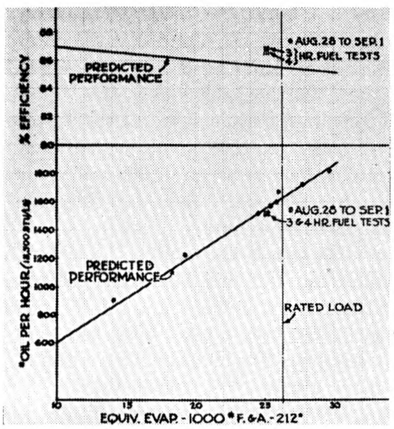

Fig. 13 Boiler-Performance Curves

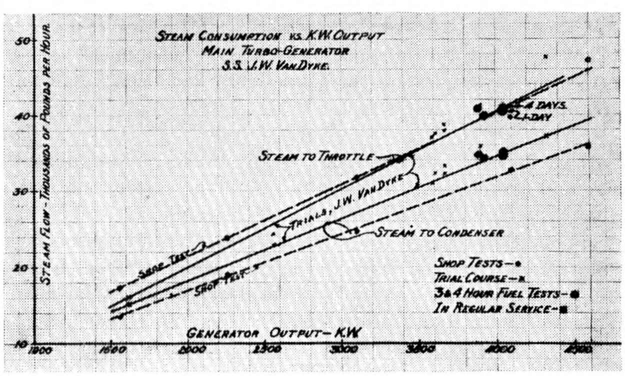

Fig. 14 Steam Consumption Vs. Kilowatt Output of the Main Turbogenerator

Since these tests were made, the results under normal operating conditions have shown that these efficiencies are easily maintainable. Data taken from the log of the homeward (loaded) leg of voyage No. 12 show that an efficiency of 87.3 per cent was maintained for the four consecutive full days of this run. We believe that such efficiencies can only be obtained and maintained by the almost perfect combustion conditions made possible by automatic combustion control.

Turbogenerator. The performance characteristics of the main generator are shown by the curves in Fig. 14. The two pairs of curves show a comparison between the shop tests of the turbine and the trial results on the S.S. *J. W. Van Dyke*. In studying these curves, it should be borne in mind that the figures for the trial runs were taken over periods varying from 4 to 10 min and that the amount to the condenser was calculated from the amounts of stage bleed which in turn were calculated from the rise in feedwater temperature. However, there were sufficient data taken during the 3- and 4-hr fuel tests to permit determination of the generator performance during these runs and, in addition, points obtained by averaging the four full days of voyage No. 12 of the *Van Dyke*, as mentioned previously, are also shown. In view of the close agreement of all of these points, it is believed that the curves, as shown, are a fair representation of the turbine performance.

Table 4 gives a summary of expected results compared to those actually obtained. There are two columns under each heading, the first giving the pounds per hour, the second the amount of heat added or removed at each point in thousands of Btu per hr. Below these columns is also shown the shaft horsepower (by torsionmeter), the pounds of total steam generated for all purposes per hour per shaft horsepower, and the pounds of 19,000-Btu fuel oil used per shaft horsepower.

Conclusions

For the purpose of our conclusions, we are selecting from Table 4 the figures for four days of voyage No. 12, as we believe these to be the most accurate data. From these it will be noted that the consumptiom of fuel per shaft horsepower for propulsion is 0.511 lb and this figure is to be compared to 0.510 lb predicted; however, it can be seen from Table 4 that consumptions as low as 0.502 have been obtained. We, therefore, think it fair to say, in consideration of all the data, that the power plant in its entirety has met the economy of the design.

It will be particularly noted that in arriving at the foregoing figures from Table 4 the portion of the fuel required for auxiliaries has not been included. The reason for this is that, generally speaking, all quoted figures for engines and other types of prime movers are on a shaft-horsepower basis, due to the fact that no two owners have the same requirements for auxiliaries for a given type of vessel. Therefore, it would be misleading to compare unless we can compare like things, and so the foregoing fuel figures are based on the propulsion requirement only.

It is believed that the foregoing will reveal power-plant efficiencies not previously approached in this country in marine installations, and that when one considers the price differential between the fuel burned in these vessels and the fuel required by Diesel engines, there should no longer be controversy relative to the most efficient type of power for this size of power plant. It is altogether fitting that the results of the engineering disclosed in this paper be presented before this Society, because it is only an adaptation of well-known principles, designs, and developments previously published through the A.S.M.E. that have been applied here. It is the earnest wish of the author that more study be given and more interest shown in ship propulsion by A.S.M.E. members in the future, so that the latest developments in power-plant equipment may be available to the marine field.

Acknowledgments

It is only fitting, at this time, that the author express his appreciation and admiration for Mr. J. W. Van Dyke, chairman of the board of The Atlantic Refining Company, for his extreme foresight in fathering this forward development, for his countless contributions to the engineering of these vessels, and for his continued patience and encouragement given the author; to Mr. R. C. Tuttle, manager of transportation of The Atlantic Refining Company, for his tolerance and cooperation; to the members of the engineering and construction department of The Atlantic Refining Company for their unfailing cooperation, loyalty, and devotion for the success of these ships; and to the suppliers of equipment, as a group, who have never tired of trying to make their equipment the best possible.

Discussion

E. G. Bailey.[2] The excellent performance of the tanker *J. W. Van Dyke* in regular service is a tribute to the farseeing and detailed research Mr. Goldsmith gave to every item in the power equipment of this vessel before the keel was laid.

In discussing the automatic control, the statement is made of the trials, "At low loads the boilers are on hand operation, hence the spread between the air-flow and steam-flow records." This was true at the time the steam-flow air-flow charts were made, but it has been standard practice for the last few months to continue on the automatic control when maneuvering in pilot waters, as well as in the open sea. The automatic control properly adjusts the fuel oil and air flow to meet the demand for steam. Burners are cut in and out for the major steps for large variations in steam demand, with the automatic control properly adjusting fuel and air within the oil-pressure range, which gives efficient atomization.

Mr. Goldsmith mentions the fact that one reason he selected the single vertical-pass gas-flow-type boiler was the entire absence of gas baffles which this design assures. Of equal importance from a maintenance and external corrosion point of view is the entire absence of shelves or ledges for soot to lodge. Any such ledges or other soot pockets are always potential fire or corrosion hazards with modern boiler-fuel oils with their frequently high sulphur content and minute particles of carbon which are frequently impossible to burn completely, even with relatively hot furnaces.

The use of air heaters as the medium for reducing the temperature of the gases well below that of the boiler tubes, not only assures desired high efficiencies with a minimum of high-pressure parts, but also is an important factor in providing optimum combustion conditions at all rates of operation.

The point is brought out in the paper that, with the type of boiler selected, prompt plugging or renewal of boiler tubes may be made without loss of ship's time waiting for a boiler to cool down enough for the drums to be entered. This is a particularly important feature in ships fitted with only two boilers, where the outage of one for a day or so may seriously affect the ship's ability to maintain her schedule.

Practical experience together with extensive tests and research work on both navy and merchant-marine installations have proved that, when burning bunker fuel oil, excessively low furnace-release rates have no commercial or thermal advantages. Mr. Goldsmith is to be congratulated in not handicapping his ship by utilizing excessively large furnaces under the boilers in the *J. W. Van Dyke.* The consistently excellent combustion results obtained in this installation serve to emphasize the commercial advantage of not wasting valuable floor and cubic space in the ship with unnecessarily large furnaces.

David Dasso.[3] The writer wishes to comment on the statement in the fifth paragraph, namely, "In other words, for horsepowers to 3000, the Diesel engine is an undeniably economical prime mover, but above that size, doubtful, especially in view of the advent of the high-pressure, high-temperature steam turbine." It is my belief the author is stating there the truth, the whole truth, and nothing but the truth, but only as applied to the steamship *J. W. Van Dyke,* operating on comparatively short runs between points in which cheap residual oils of the type called bunker fuels are available.

Beyond these restrictions, the author's statement is hardly in accordance with present-day experience. A large percentage of ship construction in all foreign countries is Diesel propelled. Many large navigation companies which previously bought nothing but steamers are buying more and more Diesel ships. There must be a reason why large Diesel ships in horsepowers ranging between 3000 and 30,000 are being ordered in increasing numbers. Are the purchasers making a mistake, according to Mr. Goldsmith's statement? They are not, and the reasons are fairly simple to elucidate.

The choice of the best type of propelling power depends on many factors. It is generally admitted today that the fuel consumption of an up-to-date Diesel plant does not exceed 0.38 lb per shp per hr, against a minimum under the best conditions of 0.5 lb per shp per hr for steam. While it is true that in some ports the steam vessel may avail itself of very cheap bunker fuel prices, this does not hold true for all ports of the world. Especially in the Far East, it will be found that the differential between the price of Diesel oil and the price of bunker oil is much less than it is on the Atlantic Coast and in the United States. The saving in fuel consumption in a vessel of the up-to-date type with a speed well in excess of that mentioned in this paper is very considerable and, taken on a run of 6000 miles or more gives to the motorship an additional freight-carrying capacity which in many cases makes this advantage decisive.

The best proof of this is the case of the large Dutch shipping companies which are building simultaneously steamships and Diesel ships, the former being allocated for medium runs between ports with favorable bunker-oil prices, and the latter for long runs between ports in which the differential in price between the two fuels is not great.

The foregoing conclusions have been affirmed in this country. Several years ago Robert L. Hague of the Standard Shipping Company, presented a paper[4] before The Society of Naval Architects and Marine Engineers discussing the subject in a similar manner. His conclusions were that, for service along the Atlantic Coast, the steamship was more economical, and for service in foreign waters the Diesel ship had been found more advantageous.

R. H. Davis.[5] In the design of any tanker, determining the type and arrangement of the cargo pumping system is a matter directly involved with the selection of the propulsive drive. The ideal condition from a fire- and explosion-hazard standpoint is a pump room, divorced from the engine room and located amidship, in which there are no wiring, electric current, or steam. With the pump room located amidship and no power available this immediately limits the selection of pumps to the vertical type, with the driving elements located on the upper deck.

The so-called deep-well type vertical centrifugal pump lends itself admirably to such an application. While a unit of such type had never been used previously for cargo pumping in a tanker of this size, it was selected in this case because of its many advantages.

The units are driven by 300-hp vertical explosion-proof motors, located on the upper deck, with the entire thrust load of the unit carried in the upper motor bearing, thus permitting complete freedom to the long length of intermediate shafting, cover pipe, and drop pipe. The pump bowls themselves are located at the bottom of the pump room in the so-called suction wells or boxes. The outside drop pipe is connected to these suction boxes through a flexible expansion joint, which also in-

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[3] Vice-president, American Locomotive Company, Diesel-Engine Division, New York, N. Y. Mem. A.S.M.E.

[4] "Performance of Standard Oil Tankers," by Robert L. Hague, Transactions of The Society of Naval Architects and Marine Engineers, New York, N. Y., vol. 40, 1932, pp. 357–368.

[5] Assistant manager, Marine Division, Worthington Pump and Machinery Corporation, Harrison, N. J.

corporates stops to prevent too great a movement. The drive shaft operates in a cover pipe which is filled with oil, supplied by an automatic lubricator above. The cargo flows between the cover pipe and the outside drop pipe and is discharged at the deck above. The lower portion of the drop pipe and cover pipe is made in the form of a split section so that the pump may be inspected or dismantled, if necessary, without disturbing the intermediate shafting above.

The important element in the ability of a ship to discharge its cargo rapidly is the method of stripping the tanks. In the case of the *J. W. Van Dyke*, no separate stripping pumps are used and yet this ship, together with its sister ship, the S.S. *Robert H. Colley* has established a record that never before has even been approached in the unloading time of a vessel of this size. The main cargo pumps are used as stripping pumps with the aid of vacuum priming pumps, which are connected to the suction boxes and so arranged that the cargo pumps are always completely primed regardless of the amount of air pulled through from the tanks. It can therefore be seen that the rate of stripping the tanks is greatly increased and the time required for stripping becomes a small proportion of the entire pumping period. Not only is the stripping time greatly reduced but this method produces amazingly dry tanks.

The characteristics of these main cargo pumps are such that the capacity increases with slight reductions in the head, in much the same manner as the conventional centrifugal pump with a relatively flat capacity-head curve, except that this capacity-head curve rises rapidly as it approaches shutoff. The shape of the brake-horsepower curve is such that it increases in practically a straight line until it reaches a maximum just before the design condition and then falls off rapidly at the outer end of the curve, thus providing an ideal horsepower characteristic for a unit which is called upon, not only to pump cargo, but to handle salt water during ballasting periods at relatively low heads. The units are powered to provide ample margin for any peak horsepowers when these units are used for ballasting when handling salt water.

The pump room is also equipped with a small 160-gpm unit of the same type as the main cargo pumps for bilge service, driven by a 3-hp motor on the deck above.

In connection with the feed system, it is interesting to note that the deaerating marine feedwater heater installed has made it possible to operate this vessel under all sea conditions with practically zero oxygen content at the feedwater as obtained by tests. This unit not only acts as a feedwater heater and deaerator, but provides approximately 2500 gal storage capacity.

E. H. Markley.[6] The writer's comments will be confined to the condensing plant which has some unusual features and which has had an excellent performance record. When the power plant of the S.S. *Van Dyke* was considered, the owners did not issue definite or detailed specifications for the condensing plant. The condenser manufacturer was requested to submit his recommendations on the best possible plant to meet the operating conditions, with the understanding that the final design would be determined after owner and condenser builder had studied all of the conditions; efficiency, reliability, space, weight, details of ship design, etc. The idea was to install the plant best suited to the operating requirements and the machinery to be served by the condenser. Results have proved the effectiveness and soundness of this cooperation.

As Mr. Goldsmith mentioned, the main condenser is of the divided type, having a total surface of 4475 sq ft, rated at a capacity of 35,900 lb at 28.44 in. vacuum, when circulated with 6775 gpm of cooling water at an inlet temperature of 76.8 F. It will be noted that these proportions take advantage of the benefits secured from the patented deaerator in that a 7.5 fps water velocity through the tubes, with its attendant economies, could be selected without fear of abnormal tube corrosion or erosion. So far there has been no sign of tube corrosion or erosion on the *J. W. Van Dyke*, which certainly justifies the installation of the deaerator. Another result of the use of the deaerator is that all condenser tubes are filled with solid water, thus raising the over-all heat-transferring capacity of the condenser.

It should be mentioned that the condenser tubes incorporate inlets belled into a hydraulic radius; water boxes are streamlined and proportioned for direct flow at exceedingly low velocities.

Another interesting point, particularly as it affects spare parts, is that the condenser-tube diameter and length are the same for the main and auxiliary condensers. The main and auxiliary air ejectors employ the same size elements.

Recognizing that a frequent and preventable cause of poor vacuum is air leakage into the condenser, the owner took precautions to prevent such leakage. The condenser is bolted directly to the turbine exhaust flange without an expansion joint, thus eliminating one vacuum joint. The expansion strains are taken up by providing a spring mounting for the condenser. Air ejectors are located on the condensers adjacent to the air offtake connections, thus eliminating the customary long air-suction pipe with its multiplicity of vacuum joints.

The benefits, derived from the careful engineering consideration given the problems surrounding the condenser installation, are evident in the following performance data which are average for one of the latest voyages: Steam load, lb per hr, 34,600; vacuum, in. Hg, referred to 30-in. barometer, 28.6; sea temperature, 76 F.

J. J. Nelis.[7] While this ship is an example of many progressive hull and engineering developments, there are some features of its power plant that differ from many of our newer ships. It has the highest pressure and temperature in the American merchant marine. The United States Navy is using approximately the same pressures and temperatures in some of its new ships. As regards pressure and temperature, the marine man is naturally conservative, as his equipment is confined in a small space, where failure of the pressure parts will endanger human life. That is one of the reasons why he has been slow to install high pressures and temperatures.

The straight-tube-type boiler, installed on this ship, has been in use for a great many years, both ashore and afloat. In the new ships now being built for the American merchant marine and for the Navy, more than 80 per cent of all the boilers installed and on order are of the bent-tube type as this type fits into the limited space conditions aboard ship to better advantage than the straight-tube type. The final-stage feedwater heater of the countercurrent flow type, commonly called an economizer for want of a better description, has also been installed on over 80 per cent of the new merchant-marine and naval vessels. The arrangement of the boiler room in this ship, requiring several frame spaces which could be used for additional cargo space, is different from other recent American tankers, where the boilers are placed aft of the turbines on a flat, thus saving the fore-and-aft space required for a separate boiler room.

Bent-tube boilers, coupled with economizer installations recently installed in new merchant and naval vessels, have shown stack temperatures averaging close to 280 F. These average

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[7] Vice-president, Marine Department, Foster Wheeler Corporation, New York, N. Y. Mem. A.S.M.E.

temperatures are lower than on the few new ships fitted with air heaters. There is an increase of nearly 2 per cent in the boiler efficiency, due to the lower stack temperatures with economizer-fitted boilers.

Modern ships are fitted with closed feed cycles which give an air-free feed. With rolled-tube condensers and air-free feed-water, few, if any, boiler repairs are required in the newer vessels. The bent-tube boiler eliminates the necessity for long repair periods, as it has only two gasketed joints for the manhole plates, as against the many small gasketed joints of the straight-tube-type boiler.

The boiler shown gives a liberation rate of 66,000 Btu per cu ft of furnace volume at full power. It would be interesting to learn what the Btu release is when the ship is washing holds at sea. In many modern tankers of from 3000 to 4000 shp, when cleaning the holds at sea, the steam required for cleaning tanks is equal to that for the power plant. Consequently, the Btu release in the furnace is then more than double the full-power conditions, due to a slight drop in boiler efficiency at the higher ratings.

With a straight-tube boiler and convection superheater to obtain superheat control requires considerable extra control equipment and a desuperheater in the drum. Marine drums are necessarily small. It is desirable to provide adequate superheat control without a drum desuperheater. A straight-line temperature curve may be obtained with the by-pass as shown in the paper, but a control of superheat at any temperature above saturation can be obtained without the complications of the desuperheater and the extra piping shown.

It is desirable to bring a marine turbine into service with a minimum temperature rise and to increase the temperature as the turbine is slowly warmed up. This requires complete superheat control which is now being used on other tankers.

In addition to complete superheat control, combustion control is now becoming standard on American ships. The new wide range oil burners, coming into use in conjunction with combustion control, give a steady efficiency without manual oil-burner operation both at sea, when maneuvering, coming in and leaving port, and when lying at the dock in port. Combustion control at sea is not so important on a ship as in a land plant, as a ship is essentially a base-load plant when at sea, the variation in steam demand being minor changes in some of the small auxiliaries. Many ships at sea are operating just as efficiently without combustion control as with it on account of their steady base-load conditions.

In connection with combustion, it is not necessary to design furnaces for high temperature to burn the fuel, particularly in marine work where the average fuel is oil having about 12 per cent hydrogen. Marine furnaces are small and instead of being heated by additional heat from the air heaters, they require cooling. In this case, the air is heated and additional heating surface is installed in the sides of the furnaces for cooling which is hardly logical.

There may be maneuvering and other advantages in an electric-drive arrangement. An interesting comparison of the heat-balance data, Table 4, of the paper can be had with that of a paper entitled "Heat Balance Calculations for Marine Steam Plants," presented at the 1938 annual meeting of The Society of Naval Architects and Marine Engineers, by A. S. Thaeler and D. C. MacMillan, where for approximately the same steam pressure and temperature, practically an equal fuel rate per shaft horsepower is shown with a mechanical gear drive and the same general heat balance.

The condensing system has several unique features. The air-removal chamber is new and may do the work expected of it. A well-designed marine condenser, having 70–30 per cent cupronickel tubes, properly designed water boxes, inlet and outlet piping, will operate without serious air corrosion for many years, if not for the life of the ship. The division of the condenser is an idea that has been tried many times and, for ships operating in tropic waters, has considerable value. It is not installed in many ships and has not yet become standard in marine practice.

In the past, marine operating engineers have seldom been technically trained men, but more of the mechanic class. Today with living conditions aboard ship greatly improved and with high-pressure, high-temperature, high-efficiency plants demanding expert attention, engineers of ability, many of them college-trained, have been attracted to the merchant-marine service. Eventually we may expect the staffs of our ships to be composed of high-grade engineers. Time only will tell which of the new and progressive ideas tried out in this ship are important enough to become standard in the future. There are other tankers recently built incorporating progressive ideas and for the pressures and temperatures used give a comparable fuel consumption per shaft horsepower. Comparisons of this with other new developments will in the end lead to better marine power plants.

Mason S. Noyes.[8] The extent of the engineering information in this paper is considerable and of great interest and value to those who are concerned with similar design work on government vessels.

The most notable items in the design are: (*a*) The slight step-up in average sea speed over present United States practice; (*b*) the selection of electrical speed reduction in place of the customary mechanical method between turbine and propeller; (*c*) the very short discharge period permitted for the ship's cargo; (*d*) the use of motor-driven cargo pumps; and (*e*) the employment of main steam conditions which, although no longer unusual for power plants ashore, are quite beyond the ordinary in present-day merchant vessels. Only in the merchant fleet of the progressive Germans can be found equal or higher steam conditions utilized for new ships.

Nevertheless, the boilers reflect conservatism, as well as the dictates of maintenance cost and time indicated by the author. The heat release, presently available in reliable marine boilers under the same steaming conditions, is several times larger than that quoted for the boilers on the *J. W. Van Dyke*. The $^3/_4$-in. (or more) thickness of the main steam piping seems definitely excessive and provides an element of danger. Such stiff piping imposes higher expansion thrusts on boiler and turbine connections, unless a more extensive piping arrangement is used to increase flexibility to the proper degree. The resulting low stresses reduce the creep rate below the neutral (or flat) value so that actually a decreasing creep rate may exist. Information is requested as to whether any "cold-spring" or pull-up was used in installing the main steam piping. What type of gasket and what gasket material were used? What is the actual pressure drop between superheater outlet and the main turbine?

The use of 2300-v cargo-pump motors is noted, together with the means provided to replace the usual stripping pumps and piping and to protect the 2300-v cables along the weather-deck gangway. Were any other special precautions taken to protect such cables from the effects of free water?

In his conclusion, the author mentions the very low fuel consumption obtained. This is due to the excellence of the heat-balance arrangement in rejecting the minimum amount of heat to the sea and to the high efficiency of the machinery items selected. Unfortunately, Table 5 does not give complete data, so that the exact extent of motor-driven units is not revealed. On the other hand, the effects of an advanced steam pressure and temperature

[8] Engineer, Design Division, Bureau of Engineering, Navy Department, Washington, D. C. Mem. A.S.M.E.

are reflected in the low machinery weights and spaces and the correspondingly high deadweight displacement ratio.

H. L. SEWARD.[9] With the experience of the author and his organization in high-temperature, high-pressure apparatus ashore, it was no extraordinary effort for them to decide to use the steam conditions, standard apparatus, and modern designs, which make these tankers so important and commendable at this time. Economy is important but dependability is more important in these vehicles, which are but a part of the machinery and equipment necessary to carry on the principal business of the company.

Those of us, whose sea experience goes back to the days when almost all of the auxiliary pumps were beam-connected onto the main engine, recall what an event it was when we got under way and started the main engine with its attached boiler feed pumps, air pump, and bilge pumps. On these tankers with a central electric-generating plant, the application of power to the propeller motor is but an incident in operation no more to be considered than the application of power to the cargo pumps or other ship's services. Aboard one of these vessels for a short trip, the writer noticed many details of design which deserve favorable comment, although not all were mentioned in the paper: Accessibility of machinery for inspection and repair; quarters for officers and men; location and type of electric cables; location of cargo-pump motors and controls; release of entrained air in circulating water; position of all motor starters on a high and protected level in the engine room; in addition to the well-known Btu savers, described in the paper, which if misapplied, may save some thermal units at the expense of over-all cost. With three fourths of the total weight of the loaded vessel as pay load, the marine industry will appreciate the achievement made in design and should watch the operating results of the next few years with great interest.

W. F. SCHULTZ.[10] The marine industry is particularly interested in this paper, since the author not only has described in much detail the machinery installation but has also presented the highly important propulsive-design information which, in the case of privately owned ships, is somewhat rare. It is noted that, in comparing the propulsive performance predicted by the Hamburg tank, and that by the Washington tank, there is considerable divergence, although the predicted power-revolution curves are practically coincident for both tanks, and with the actual ship performance on trial. It is noted that the actual ship shaft horsepower is about 16 per cent higher than the Washington tank prediction at the designed speed of $13^1/_4$ knots. This is not unusual since the trial course off the Delaware capes is not as suitable for comparing model test data with actual ship performance as is the Rockland course. It is probable that sea and wind conditions together with depth of water and the effects of current have increased the power requirements above ideal trial conditions.

In discussing the power plant, however, there is considerable difference of opinion as to the most economical installation possible, and it is in this connection that the writer wishes to discuss the paper.

The author is to be commended for his selection of steam conditions. It is the first instance in which an American-built merchant ship has been fitted with steam pressures and temperatures above 400 lb and 750 F. These conditions are even somewhat unusual in shore plants in the same power range, although there are several. To the progressive engineer, however, the attractive features of higher pressures and temperatures make such adaptation worth while. It may be interesting to consider the economies possible through their use.

Table 6 gives a brief summary of the pertinent features of four hypothetical installations, each having different steam conditions; and each progressively higher. In this case the installation is for 6000 shp, employing multistage feed heating, all-electric auxiliaries, double-reduction-geared turbines, and boilers fitted with air heaters and economizers. An attached generator, driven from one of the low-speed pinions, furnishes the electric power for sea operation, similar to the equipment fitted in fourteen ships built by the company with which the writer is associated.

TABLE 6 SUMMARY OF STEAM CONDITIONS, TYPICAL PLANTS

Installation	A	B	C	D
Pressure, superheater outlet lb per sq in., gage	460	600	900	1200
Temperature F, superheater outlet	750	850	900	950
Feed temperature F, economizer inlet	318	332	365	400
Feed heaters, number of stages	3	3	3	4
Steam flow to condenser, ratio	1.0	0.91	0.83	0.80
Boiler heat output, ratio	1.0	0.97	0.91	0.88
Fuel, lb per shp per hr, all purposes	0.588	0.568	0.534	0.517
Fuel rate, ratio	1.0	0.967	0.908	0.88

It will be noted that the machinery installation on the *J. W. Van Dyke* is in some respects similar to installation B (Table 6). There are, however, differences that make comparison with this installation somewhat difficult. Therefore, the writer has prepared Table 7, showing the differences between an installation similar to B and that of the *Van Dyke.* Both plants have the same steam conditions and will deliver the same shaft horsepower, but for the proposed installation the writer has made certain substitutions.

The main propulsion machinery is of the cross-compound turbine double-reduction-geared type, without attached generator. The boiler plant consists of two 2-drum units fitted with air heaters and economizers. The feed system is of the three-stage type, with surface-type heaters and deaerating-type surge tank. The main feed pump is electric-motor-driven, as are the feed-heater drain and booster pumps. The electrical load requirements are provided for in two 350-kw, 230-v, d-c generators, one of which will carry the normal sea load at about 50 per cent rating. The two units would be sufficient for cargo pumping requirements in port. A contaminated steam system is included similar in design and purpose to that of the *Van Dyke.*

Based on the information given in the paper, the estimated fuel rate for 5000 shp, based on 19,000 Btu oil, is 0.596 lb per ship per hr for all purposes. Service performance has proved the author's estimates to be very accurate, speaking well for his own ability, and for the reliability of information from the suppliers of his machinery.

We are accustomed to base our performance on 18,500 Btu oil, which is a fair average value for the fuel supplied today. Referred to this value the fuel rate of the *Van Dyke* is about 0.612 lb per shp, say 0.61 lb per shp per hr for average service conditions.

Comparing this performance with an installation similar to B in Table 6, we find that a gain in economy of approximately 7.5 per cent is possible. Analyzing this gain, it is found that the reasons are generally as set forth in Table 7.

In explaining the apparent reasons for the increased economy, noted in Table 7, the writer wishes to discuss the various principal items of machinery in their order of importance.

The cross-compound turbine, as furnished by the principal machinery builders, either impulse or combined impulse-reaction

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[10] Engine Department, Federal Shipbuilding and Dry Dock Company, New York, N. Y.

TABLE 7 COMPARISON OF OPERATING EFFICIENCIES

Installation	*J. W. Van Dyke* A	Proposed B	Increased or decreased[a]
Turbine efficiency, per cent	75.6[b]	77.7	+2.7
Transmission efficiency, per cent	93.4	97.5	+4.3
Auxiliary efficiency, per cent	..	..	—0.5[c]
Boiler efficiency, per cent	87.0	88.0	+1.1
Total increase or decrease in economy			7.6

[a] Increase or decrease referred to A.
[b] Estimated from test data given in paper.
[c] Effect of operating ship auxiliaries at sea from main propulsion unit.

type lends itself particularly well to obtaining relatively higher efficiencies than the single-cylinder unit such as used on the *J. W. Van Dyke.* Higher speeds are possible, smaller shafts and smaller-diameter wheels can be used, thus reducing internal friction and bearing losses, and generally lighter units are possible. In the paper, the weight of the main propulsion unit, complete, is 211,000 lb. Recent 6000-shp turbine proposals for cross-compound 18-stage double-reduction-geared units varied from 174,000 to 180,000 lb, a saving in weight of more than 30,000 lb. It is obvious of course, that this difference in weight would indicate a substantial reduction in cost. This difference is more pronounced based on the evaluated cost, in consideration of efficiency due to higher speeds and greater number of stages. It should be noted, however, that the reduction gears proposed were of the modern single-case type, and not the extremely heavy three-case type of five years ago.

A modern double-reduction gear of the single-case type, with two bearing pinions, and tooth pressures of the order of 60 lb per in. of face per in. of pinion diameter will give a mechanical efficiency of not less than 97.25 per cent and in most cases 97.5 per cent or better, as compared with a combined efficiency of from 93 to 93.5 per cent for generators and propulsion motor. In addition, the space required for the geared-turbine unit is less than that of the electric drive for the same power.

In discussing boilers, the writer will not attempt to criticize the type used. It is generally a matter of opinion as to the type best suited for the purpose. However, it is believed that an economizer in combination with the air heater may have been used to advantage and the efficiency raised to 88 per cent, at a relatively low increase in cost, and with no appreciable increase in weight.

It is the writer's opinion that wider use of the completely water-cooled furnace, as is rather common practice in shore power plants, will prove of considerable economy to the marine field. The elimination of brickwork, especially in furnaces designed for high rates of heat release, will reduce furnace maintenance to a minimum.

The author is to be congratulated on the method of temperature control applied in this installation. It is believed to be superior to other forms, and lends itself particularly well to shipboard use. We have contemplated a similar arrangement for use in high-temperature installations for desuperheating steam to the astern turbines during maneuvering. We have introduced feedwater into the main steam line for this purpose in the past, but this method is relatively crude compared with the excellent arrangement used on the *Van Dyke.* By this method, the one principal disadvantage of the geared-turbine installation may be overcome without great difficulty.

The application of automatic combustion control is of great interest. The writer fully concurs with the author that the use of higher pressures and temperatures will further increase the use of this equipment beyond its growing application today. It is now possible to obtain from at least two sources in this country, a wide-range oil burner that will greatly augment the value of the controls for all-purpose operation. A recent installation in a tanker provides us with a very instructive case in point; the ship was able to get under way, maneuver, drop anchor, and perform all the necessary tasks including docking without use of towboats, with all burners in use and under full automatic control.

Concerning the type of current used by auxiliary drive, the author had available the main propulsion generator for supplying power at sea and, with the selection of this type of drive, the best arrangement would be the simplest, namely, alternating current at a standard voltage. However, with geared turbine drive it is believed that direct current is more suitable since the problem of speed regulation is relatively simple. It is questionable in my mind, when the cost of the controls necessary to provide speed regulation for fuel and feed pumps, forced-draft fans, and cargo pumps is included with motor costs, whether there is any appreciable difference in cost. The space required by a-c controls is considerable and would probably minimize the gain in space inherent with smaller sized alternating current motors.

Among the interesting features of this ship are the cargo-pumping arrangements. Excellent judgment has been used in adopting explosion-proof motors, since tanker practice requires the ultimate in safety. This ship is engaged in the gasoline trade and handles great quantities of this fluid at high pumping rates, requiring extreme care. All the demands of this service seem to have been amply met. It would be interesting to obtain the author's reaction to having this same type of pump installed in the pump room below decks, and providing forced ventilation from fans on the deck, with elevated intakes.

Author's Closure

In his comments Mr. Bailey has referred to the combustion-control features of the *Van Dyke.* The record of the use of the automatic combustion control on this vessel followed exactly the predictions made for it. At first it was only used when under full or nearly full load and not in pilot waters. The next step was to leave the fuel under automatic control at all times and to regulate the air supply by hand at low loads or when maneuvering. As stated by Mr. Bailey, the automatic control is now used at all times but, at low loads or when maneuvering, the air flow is set up to avoid the possibility of smoking when the load is suddenly increased.

In reply to Mr. Dasso, the author's ideas as to the relative merits of turbine versus Diesel drive are obviously based on the conditions which he has to meet. He agrees that for other conditions the selection might have been different. It should be noted, however, that, using Mr. Dasso's figures, the ratio of fuel consumptions of turbine versus Diesel machinery is 1.32 to 1, whereas the ratio of fuel cost of Diesel to bunker C oil at present varies from 1.6 to 2.07 to 1, which is a fairly wide margin in favor of steam. It should be borne in mind also that the European preference for the Diesel is to some extent due to the fact that most of the shops and shipyards abroad are unable to build modern high-pressure, high-temperature turbines. Also it is an easier step for an engineer who has been trained on a reciprocating steam engine to change over to the Diesel type.

Mr. Nelis refers to the wide adoption of the bent-tube boiler in new merchant and naval vessels in the United States. In the author's case the lighter header-type, straight-tube boilers enable more cargo to be carried than if multidrum, bent-tube-type boilers were installed, regardless of space arrangement. This is due to the fact that, while the *Van Dyke* has a tank capacity for 18,600 tons of 60 deg Bé gasoline, the cargo allowed is only 17,125 tons with a deadweight tonnage of 18,105.

Clean and uncorroded boilers are admittedly possible with proper feed preparation but this cannot always be assured, as witnessed by the recent difficulties with the boilers of a modern tanker, which might have been avoided if the tubes could have been inspected. No difficulty has been experienced in keeping handhole covers tight. It is the author's belief that there is but slight difference in the number of gasketed joints between a header-type boiler with an air heater and a bent-tube boiler with an economizer.

The relative boiler efficiencies obtainable with an economizer as compared to an air heater depend entirely upon the size of the air heater. On a number of recent ships, stack temperatures as low as 280 F have been obtained with air heaters. During the fuel-consumption trials, the stack temperature on the *Van Dyke* averaged 323 F. In this connection it should be noted that a gain in boiler efficiency may be made at the cost of a greater loss in over-all efficiency, as is witnessed by the figures given in the paper by A. S. Thaeler and D. C. MacMillan, to which Mr. Nelis refers. It will be noted that columns A and C in Fig. 5 of that paper compare almost identical setups, except that the boiler of column A has an economizer only, with a feed temperature of 235 F. In contrast the boiler of column C has an air heater and an economizer which permits three heating stages instead of two with a feed temperature of 300 F, resulting in a fuel reduction of 110 bbl per hr, or 2.3 per cent.

The heat-liberation rate in the *Van Dyke* boilers (66,000 Btu per cu ft per hr) is extremely conservative in the light of present practice (note Mr. Noyes's comments). Even if doubled it would still be well under what has proved to be possible. This, of course, depends on the burners used and the arrangement of the combustion space and surrounding cooling surfaces. In general, it is not the combustion space but the capacity of the downcomers which limits the capacity of a boiler.

With regard to superheat control, the author fails to see that there is essentially any greater complication in controlling a bypass valve to the desuperheating coil than there is in controlling another oil burner.

No difficulty has been experienced in obtaining all necessary control of steam temperature when starting the turbines on the *Van Dyke* cold.

It is agreed that, for perfectly steady loads, it is theoretically possible to operate efficiently without automatic control. Evidently this generally is not done, since as Mr. Nelis states: "Combustion control is now becoming standard on American ships."

The author is in agreement with Mr. Noyes, concerning the excessive thickness of steam piping, but is happy to say that the Bureau of Marine Inspection and Navigation has recently issued a ruling which will permit advantage to be taken of the higher creep strength of carbon-molybdenum tubing on the third ship now under construction. In this case the thickness of the 5-in. main will be reduced from $^3/_4$ in. to $^1/_2$ in., approximately.

In answer to specific questions by Mr. Noyes, the following information is given:

Piping was "cold sprung," approximately 50 per cent of the expansion.

Gaskets were originally of soft (Armco) iron, but these were found to be unsatisfactory due to the temperature changes incident to shutting down at weekly intervals. "Flexitallic" gaskets have successfully replaced the solid soft iron.

The pressure drop from superheater outlets to turbine throttle is 10 to 15 lb at full load.

Cables of 2300 v on the catwalk are 106,000 cir mils with 2400-v lead and armored, basket-weave marine insulation. They are fastened to the inner surface of the catwalk side by lead-covered brass clips and covered by a brass housing bolted to the side panels.

Table 2, given in discussion by Mr. Schultz, compares a hypothetical power plant with that on the *Van Dyke* and shows a gain of 7.6 per cent in efficiency, partly due to the assumption that higher turbine and boiler efficiencies might be obtained. This is no doubt possible but the author's original investigations, preparatory to selecting the equipment for the *Van Dyke*, led to the conclusion that the extra cost of so doing would n t be justified in a plant of this size. The remaining gain is based on the higher transmission efficiency of a gear drive. Granting this gain to be possible, it was felt that the advantages of the electric drive, as outlined in the paper, more than offset this difference. However, it may be noted in passing that the oil coolers generally specified for geared installations have capacity to remove a heat equivalent of more than 5 per cent of the rated power of the unit.

Regarding Mr. Schultz' request for the author's reaction to the use of deep-well cargo pumps, having vertical motors below decks with forced ventilation, it is that this greatly increases the fire hazard, requires the added cost and complication of forced ventilation, considerably increases the work of the pumper, and only gains a reduction in the length of the shaft and cover pipe, which hardly seems worth while.

Rollcurve Gears

By H. E. GOLBER,[1] CHICAGO, ILL.

In automatic machinery the movements of the various bodies are usually obtained from the motion of a follower compelled by a driver. The driver is usually assumed to rotate at a uniform speed and the follower is given some definite motion, ordinarily a definite rotation. When the follower motion is to be a rotation at uniform speed, the driver and follower are usually connected by ordinary circular gears. When the follower motion is to be a rotation of nonuniform speed, elliptical gears are sometimes used. When the follower is to move in a more complicated manner, all kinds of devices, such as cams or planetary levers, are used. Very frequently the designer does not know of any means for producing the desired follower motion. Therefore, he is compelled to use an undesirable motion which he knows how to produce, and abandons what he considers the best.

This article discusses rollcurve gears. It shows that by using such gears, the follower can be given practically any desired motion, thereby enabling the designer to produce the follower motion desired and obtain the best condition in the new machine. The article first illustrates actual rollcurve gears installed and in use on actual machines. It discusses their durability and applicability, and gives a short history of some of them. This is to assure anyone that these gears are reliable and that their production is practical.

The article next shows the production of the "speedgraf," which is a standard method developed by the author to make the desired method perfectly definite. It is then shown how to convert the speedgraf into rollcurves which may then be drawn and tested on the drawing board. The rollcurves can then be used to obtain patterns for the gear blanks. These gear blanks then have the teeth cut into them, converting them into a pair of gears. When the driver moves at uniform speed, the follower moves in the desired motion. The result is positive and reliable.

Certain features that occur in connection with the rollcurve gears are explained. Also, to show the broad applicability of the gears, a few speedgrafs, for which gears have already been made, are shown and are commented upon.

THIS article is submitted by the A.S.M.E. Graphic Arts Division because the subject matter was developed in connection with the design of printing machinery and the applications shown are upon printing presses. However, the subject is really one of applied mechanics and is equally applicable to all classes of automatic machinery.

This article deals with means and methods for producing motions that hitherto have been either impossible or impractical. The motions now capable of production are of the greatest variety; they are produced by means that the writer calls "rollcurve" gears.

Years ago it was possible to design spur gears the axes space of which was standard. The pitch diameter of the spur gear was always the number of teeth divided by the diametral pitch. If the designer were asked to make a nonstandard axes space or nonstandard pitch diameter, he would be in trouble. Finally, however, there was introduced the hobbing machine and the Fellows gear shaper, with which the designer can readily make gears before thought to be impossible. For instance, the author has made three coaxial gears, one with 8 teeth, a second with 10 teeth, and a third with 12 teeth, all simultaneously geared with a common rack. The previous cramping limitations have vanished.

Again in bevel gears, previously the two axes had to intersect and could not pass each other. Now with hypoid gears the axes may pass each other. The cramping limitations have vanished.

Similarly, designers have made use of cams and roller arms. For a reciprocating irregular motion, of say 30 deg, there was no trouble. For a reciprocating motion of, say, 90 or 180 deg, auxiliary mechanism had to be introduced. For irregular continuous rotation the difficulties were large and so the designer avoided them. It will be found that in the rollcurve gears, these cramping limitations are gone, and the designer is able to produce in a simple and positive manner almost any motion desired, regular or irregular. The possibility of design has automatically vastly expanded.

A rollcurve gear is one that has a rather uncommon rolling curve as its pitch line. In ordinary spur and annular gears, the pitch lines are rolling circles. Also gears have been made with ellipses as their rollcurves. Some suggestions have been made to use equiangular spirals as rollcurves. The most extensive treatment of the subject known to the author is the one by Robinson,[2] who gives a rough method to find the mating rollcurve to any given rollcurve. So, to a given rotation of the first, the second must rotate in a definite manner. No attempt is made in that book[2] to treat the accelerations, velocities, or displacements of the driver and follower.

About 1931 the author developed some methods for making rollcurve gears, and they were applied in practice. His method was not derived from any previous method, although at places there necessarily must be some resemblance since the subject "rollcurve gears" is the subject in common.

The basic difference between say Robinson's methods[2] and the author's is that Robinson assumed one rollcurve and then tried to find the other, while the author first assumed the motion of both the driver and follower and then found the pair of rollcurves at once. Robinson thus obtained a motion of the follower

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Contributed by the Graphic Arts Division and presented at the Annual Meeting of The American Society of Mechanical Engineers, held in New York, N. Y., December 5–9, 1938.

Discussion of this paper was closed January 10, 1939, and is published herewith directly following the paper.

[2] "Principles of Mechanism," by S. W. Robinson, John Wiley & Sons, Inc., New York, N. Y., 1910.

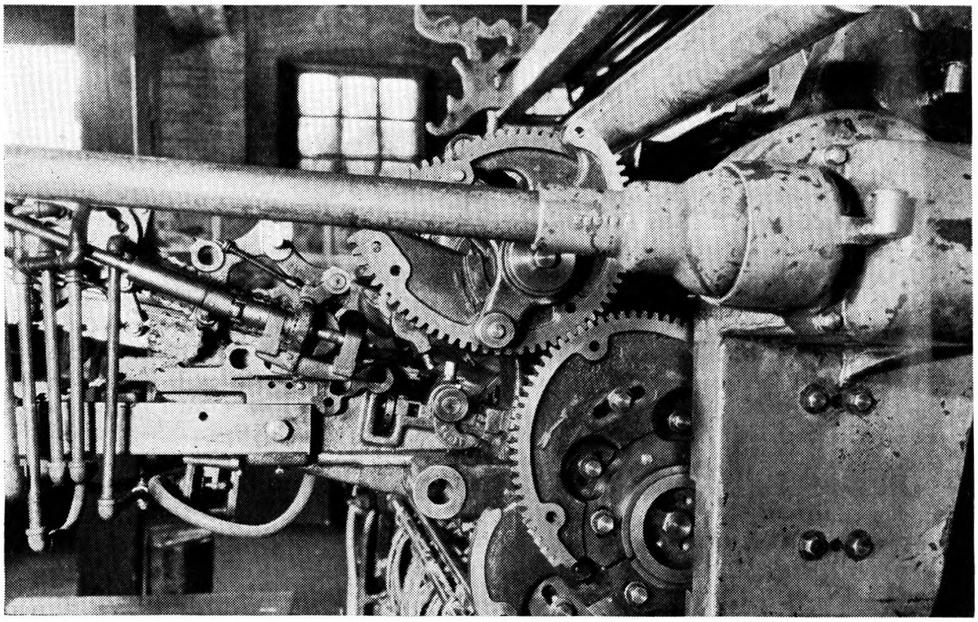

FIG. 1 THE FIRST ROLLCURVE GEARS

which depended on the properties of the assumed rollcurve. The author started with the desired motions and thereby made the rollcurves depend upon the motions. The result was that Robinson used well-known curves, say the circle, the ellipse, and the equiangular spiral. His curves were simple but his motions were complex. The author uses the circle and curves that have no names as yet. He thus obtained unusual curves, but his motions are simple. The simplicity of the motions has allowed them to be used extensively in machinery.

Observe the various illustrations of rollcurve gears. Some of them show the gears mounted upon and forming parts of machines. All are of gears for actual machines, not for models or experimental mechanisms. Many sets of gears have been made, installed, and used. They are still in use.

Immediately the questions arise: (1) How long have they been used? (2) How well do they wear? (3) How costly are they to manufacture? (4) Where should they be used? The answers in the order asked are as follows:

(1) The earliest pair was built about 1931. It is still in use. (2) They wear about the same as other gears made of similar material, cast iron or bronze. As yet there have been no replacements. (3) A first pair of gears is rather costly. They can then be reproduced quite commercially. (4) Where shall they be used? This must be left to the judgment of the designing engineer. Here he might take into consideration that by these gears, he has a tremendous broadening of the possibilities of design. He is able to produce in a positive simple manner practically any motion. Motions that otherwise are impossible or very difficult are easily made by rollcurve gears. Moreover he is able to reduce the number of links in the driver-follower chain, and thus reduce backlashes and improve register.

The author is continually called upon for more and more new rollcurve gears desired by the firm where the gears were first used. Some calls have also come in from outside.

CONSTRUCTION OF THE GEARS

We start with the driver and the follower. How these are to move depends of course on what they are to do in the machine. We assume that the designing engineer knows what he wishes to accomplish. From his knowledge he finally produces a description of the desired motion of both the driver and follower. This description the writer always finally reduces to a designer's "speedgraf."

Speedgraf. In books on kinematics we find various diagrams called "displacement diagram," "velocity diagram," and "acceleration diagram." The author refers to these as "sitegraf," "speedgraf," and "celgraf," respectively. Of these the speedgraf (or velocity diagram) is much more useful than the others and in practice he finds it is sufficient for his needs. Only very seldom does he need some additional diagrams.

In various books he has seen given a displacement diagram for the follower displacement, a velocity diagram for the follower speeds, and an acceleration diagram for the follower acceleration. Three diagrams are given. With very little training the engineer can obtain all this information at one glance upon the speedgraf only, and he needs no others. Thus looking at the speedgraf the observer finds the follower displacements, as the areas by the speedgraf; the follower velocities, as the ordinates of the speedgraf; the follower accelerations, as the slopes of the speedgraf; and the follower impulse, as the bent of the speedgraf.

From now on we shall consider the speedgraf as the fundamental graph—the start is with the "designer's speedgraf."

Speedgraf Segments. If the pair of gears is a pair of ordinary circular gears, the speedgraf is a line parallel to the speed-line zero. This is usually a horizontal line. Thus a horizontal line represents uniform rotation for both the driver and follower. When the speedgraf line is above the zero line, the driver and follower rotate in opposite directions as in a pair of spur gears. When the speedgraf line coincides with the zero line, the follower rests or dwells. When the speedgraf line is horizontal but below the zero line, the follower and driver rotate in the same direction.

It is usually assumed that the driver rotates at a uniform speed. The follower rotates at the speed and in the direction desired. When the follower speed changes, its speedgraf curve is not a horizontal. When the speedgraf curve is an oblique (straight line), the speed changes at a uniform rate. The follower motion then is like the motion of a body falling under gravity.

At first, each speedgraf was made as a polygon of oblique and horizontal lines. By such a polygon, varying motions and uniform motions were very nicely represented in a simple manner. But every cusp or point of the polygon spoiled the smoothness of the motion and caused trouble, first in the making of the gears and later in the finished machine itself. As these troubles were serious, they will be gone into more fully.

It was found that every speedgraf cusp produced a corresponding rollcurve gear cusp. This can be readily seen from Fig. 1. This pair of rollcurve gears is the first chronologically, being built early in 1931. The upper gear is the follower; the lower gear is the driver. The gears were made out of pieces of elliptic gears fastened together. Note that the follower gear segments are in several planes. The driver segments are similarly arranged but are covered over.

Note the rollers on the follower. One is clearly in sight, while the other at the top is half hidden. To cooperate with them is a cam, only part of which is visible at the lower left.

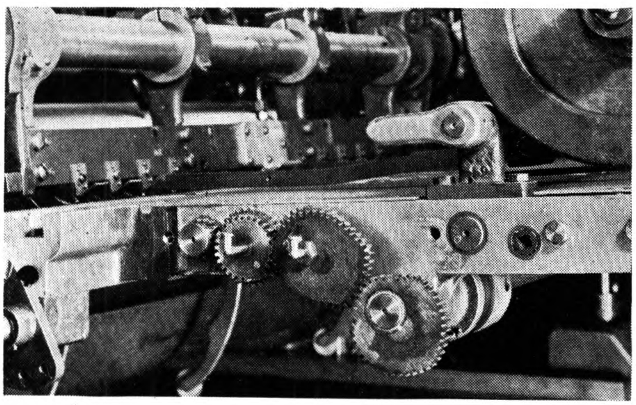

Fig. 2 Two Small Continuous Rollcurve Gears

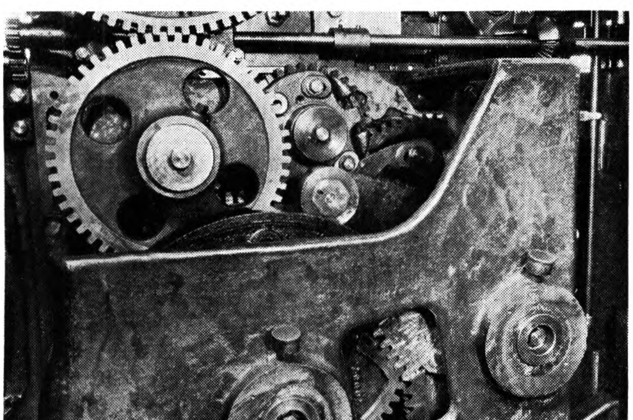

Fig. 3 Rollcurve Gears, the Driver and Follower of Which Have a Convex and Concave Cusp, Respectively

This pair of gears was made before the development of the speedgraf method. In fact, it was the difficulties encountered in this pair that drove the writer to the consideration of the speedgraf and finally to the development to be described.

As the next example, consider Fig. 2. Here the two little gears are continuous and all in one plane. They are made of bronze cast with the teeth. The lower right gear is the driver. Each gear has three cusps. This spoiled the continuity of the gear rollcurve, making the cutting of the original master gears quite

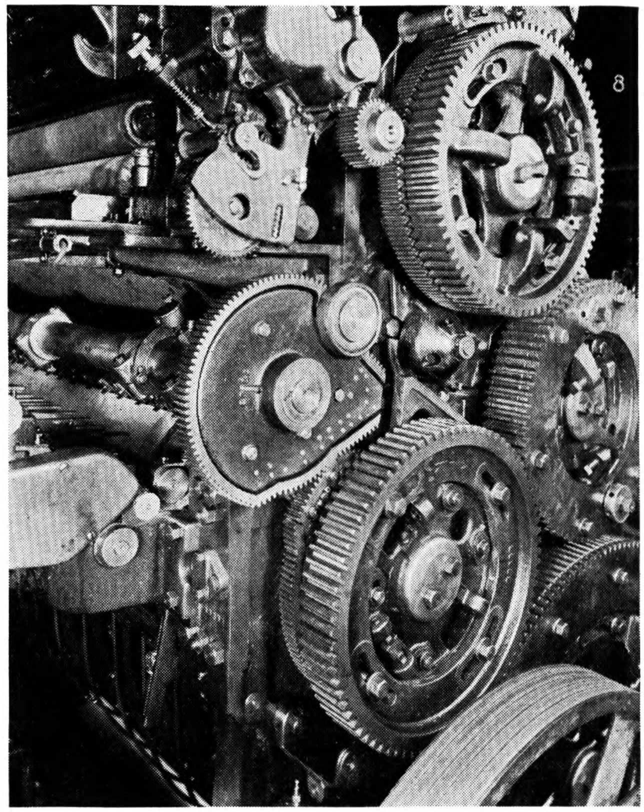

Fig. 4 Rollcurve Gears, the Teeth of Which Are All in the Same Plane

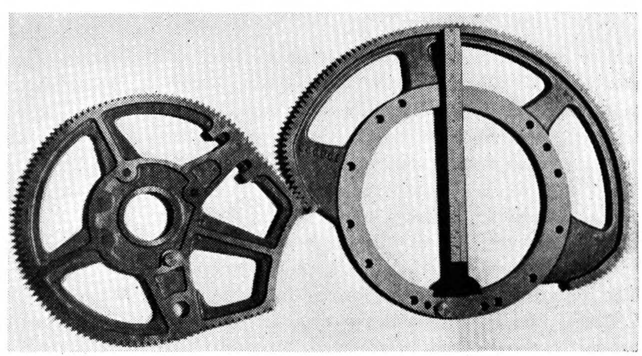

Fig. 5 Rollcurve Gears With Cusps Eliminated Which Permitted the Gears to Be Cut With a Fellows Cutter

difficult. These gears were by far the smallest, being about 3 in. in the axes space.

As the next example, consider Fig. 3. Here most of the mechanism is invisible but a portion of the driver (at the left) and the follower (at the right) are visible through the window. Note that the driver has a convex cusp and the follower a corresponding concave cusp. Here is also seen that the apex of the convex cusp has a gear space and the apex of the concave cusp has a gear tooth. This rule was found to be indispensable. Note that the follower had to be built in pieces to permit the cutting of the gear teeth. The mechanism was strong and heavy, the teeth having diametral pitch of four.

We next come to Fig. 4, in which the follower is very prominent. A portion of the driver is visible and in mesh with the follower. Note that the follower rollers have been very much enlarged. The gear segments are still made in different pieces fastened together. However, unlike Fig. 1, the teeth are all in one plane. The teeth of the driver and follower have a pitch of eight, the big gears have a diameter of 20 inches and a pitch of four. Fig. 4 shows that the driver has quite a hump.

This was the last pair of rollcurve gears to be built in that manner. Strenuous efforts had been made to devise some scheme for the elimination of the cusps, so that teeth could be cut continuously. This was finally done successfully and the gears henceforth were cut by a single Fellows gear shaper cutter. As an example, see Fig. 5. The cusps are gone, the gears are continuous all in one plane, and easily manufactured. Moreover they are accurately machined, say to 0.001 in.

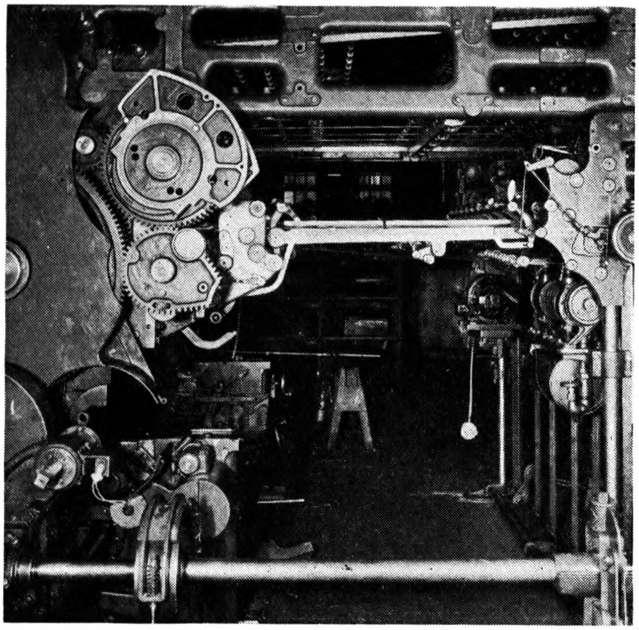

Fig. 6 Large Rollcurve Gear

There is really nothing more that is novel to show; however, Fig. 6 will serve to exhibit the extent of the use of such gears in a large machine. The big gear at the left is 36 in. in diameter.

These illustrations will serve to prove that rollcurve gears are in practical commercial use. Moreover, attention must be called to the fact that each machine shown was not an individual machine but one of a line of machines marketed.

Returning now to the speedgraf and its cusps. One reason for objecting to cusps is the consequent cuspiness of the rollcurves, and resultant difficulty in making the gears. There are, however, other objections. Thus, upon examining the speedgraf at a cusp, we see that the slope or grade suddenly changes. That is, the acceleration suddenly changes. Now machines are assumed to be built of rigid material, say, cast iron. But there is no material absolutely rigid. Every body bends when forces are applied to it. Assuming Hooke's law that "strain is proportional to stress," then every body in a machine bends an amount proportional to the stress on it. As in a machine, the forces are proportional to the square of the machine speed; therefore, in a machine the bends of the bodies are proportional to the square of the machine speed times acceleration. At every speedgraf cusp, the theoretical bending of the bodies have two different bendings corresponding to the different grades. It frequently happened that the bodies were springy and bent sufficiently to cause a misplacement of the parts, and at high speed the machine "knocked" and did not "register." All these troubles were cured by eliminating the speedgraf cusps.

So the speedgraf polygon cusp was rounded over by a filleting arc. Here many choices could be made for the arc. It could be a portion of a circle, an ellipse, an harmonic, or a parabola. The choice finally fell upon a parabola with a vertical axis. This was because in computation it was simpler to deal with a parabola, than with a circle. True, in drawing, the circle was easier, but the parabola was never drawn except very roughly. The vertical parabola won out and is the filleting curve now used.

To summarize the foregoing, the speedgraf curves used are: (1) The horizontal straight line for uniform motion of the follower; (2) the oblique for nonuniform motion; and (3) the vertical parabola either U or ∩ for uncusping the speedgraf. How these are to be combined depends of course on the machine needs and the designer's best judgment. The final result is the designer's speedgraf.

Designer's Speedgraf. The designer's speedgraf takes many shapes. Fig. 7 shows a comparatively simple one. Here we see that the speedgraf is $ABCDEFGH$. As this is a full cycle, A and H are the same point. The speed-line zero is $A_0B_0C_0D_0E_0F_0G_0H_0$, so $A = A_0$, $B = B_0$, $G = G_0$, $H = H_0$. In the speedgraf, AB and GH show speed zero, the portion $G(H=A)B$ showing a follower dwell. The horizontal DE shows a follower uniform rotation in a direction opposite to the driver. The obliques BC and FG show follower increasing and decreasing speeds. The curves CD and EF are parts of a ∩ parabola. The points D and E are therefore vertices of the parabolic arcs.

To show that the speedgraf covers a complete cycle, the column of zone angles is given. The total AH = 360.0000000 deg. The column of ordinates is given, having been determined in some manner by the designer. There are other items, such as 180 deg = basic gear space on driver, which are also given.

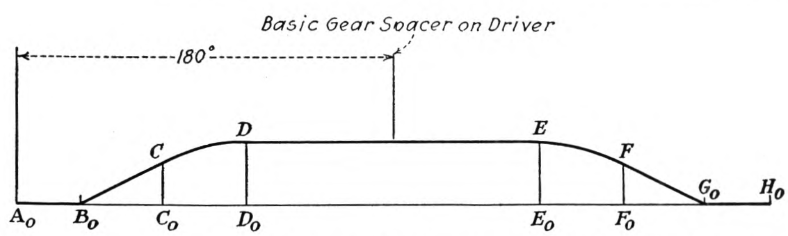

Fig. 7 Designer's Speedgraf for a Gripper Drive

A_0B_0 =	29.6388885 deg	A_0A	= 0
B_0C_0 =	40.0000000 deg	B_0B	= 0
C_0D_0 =	40.0000000 deg	C_0C	= 1.00628931
D_0E_0 =	140.7222230 deg	D_0D	= 1.50943396
E_0F_0 =	40.0000000 deg	E_0E	= 1.50943396
F_0G_0 =	40.0000000 deg	F_0F	= 1.00628931
G_0H_0 =	29.6388885 deg	G_0G	= 0
A_0H_0 =	360.0000000 deg	H_0H	= 0

Axes distance = 16.625 in.
Arm for roller = 5.75 in.
Arms angle = 134 deg
Diametral pitch = 8
Roller diameter = 120 mm
Gear cam equal steps = 1 deg
Roller cam equal steps = 1 deg

As the drive part $GHAB$ is a dwell, it is necessary to provide for it in some manner. For it, a follower tooth would have to be kept at the center of the driver axle, and this would be very unsatisfactory for several reasons. Therefore, the dwell part is made not by rollcurve gears, but by cams and roller arms. Moreover, the gear portions on BC near B, and on FG near G, are not satisfactory because they unmesh, and so the cam drive is made to extend and cover the unsatisfactory parts of BC and

TABLE 1

Rectangle A_0ABB_0	$= A_0B_0 \times A_0A$	$= 29.6388885 \times 0.0$	= 000.0000000 deg
Triangle BCC_0	$= (1/2)B_0C_0 \times C_0C$	$= (1/2) \times 40.0000000 \times 1.00628931$	= 20.1257862 deg
Parabolic C_0CDD_0	$= (1/3)C_0D_0 \times (C_0C + 2D_0D)$	$= (1/3) \times 40.0000000 \times (1.00628931 + 2 \times 1.50943396)$	= 53.66876305 deg
Rectangle D_0DEE_0	$= D_0E_0 \times D_0D$	$= 140.7222230 \times 1.50943396$	= 212.4109023 deg
Parabolic E_0EFF_0	$= (1/3)E_0F_0(2E_0E + F_0F)$	$= (1/3) \times 40.0000000 \times (2 \times 1.50943396 + 1.0062893)$	= 53.66876305 deg
Triangle F_0FG_0	$= (1/2)F_0G_0 \times F_0F$	$= (1/2) \times 40.0000000 \times 1.00628931$	= 20.1257862 deg
Rectangle G_0GHH_0	$= G_0H_0 \times GG_0$	$= 29.6388885 \times 0.0$	= 00.0000000 deg
			360.0000008 deg

FG. The required data for the cam drive are given in the lower column.

It has been stated previously that all cusps are to be removed from the speedgraf and yet there are two left, *B* and *G*. But these two occur not on the gear portion but the cam portion, and the follower speed is very low. In some speedgrafs even the speed-zero cusps are rounded out.

The designer might have been very careful. But did he make a mistake in his computation? To some extent this can be tested from the data given and implied. It was understood that while the driver made a complete rotation, so did the follower. The driver rotated uniformly and made 360 deg per machine cycle. The follower rotated always in the same direction while starting, speeding up, speeding, slowing down, or resting. But in one machine cycle it also made 360 deg. We have already stated that in a speedgraf the area represents the displacement. For the driver the area would be 360 deg × 1 = 360 deg. For the follower, the cycle displacement must be the area by the speedgraf or the area *ABCDEFGH*, which we see is equal to the values given in Table 1.

The result for the follower area we call the area angle. It should be exactly 360 deg. It is 360.0000008 deg. We assume the agreement to be accurate enough; that is, the follower rotates once for every rotation of the driver. Our first check is satisfied.

The above accuracy seems to be silly. But as we proceed, the errors that occur, because of the inability of the author to integrate accurately for the length of the arcs, reduce the above from ten significant figures to seven, which is about right to make rollcurve gears correct to 0.001 in. on an axes space of 36 in.

We next examine the tangencies. The grade at *C* of *BC*, and the grade at *C* of *CD*, must be the same. The grade at *C* of *BC* is

$$\frac{C_0C}{B_0C_0} = \frac{1.00628931}{40.0000000} = 0.0251572327$$

The grade at *C* of *CD* is

$$\frac{D_0D - C_0C}{(1/2)C_0D_0} = \frac{1.50943396 - 1.00628931}{(1/2) \times 40.0000000} = 0.0251572325$$

Again the agreement is close enough.

At *D* the parabola is tangent; similarly at *E*. We see also that *F* is symmetric with *C*. Thus far the speedgraf checks. We assume that it is correct in all details.

The Rollcurves

Having now an accepted and tested speedgraf, the next step is to obtain the rollcurves. In Fig. 8 let *C* and *F* be the driver and follower axes, respectively, with the rollcurves *V* and *W* contacting at their tangent point on the axes line *FD*. Let *r* and *R* be the driver and follower "rays," respectively. Note the word used is "rays" and not "radius," the latter term being limited to the distance from the center of a circle. Let $d\theta$ and dw be the respective small rotations of the rollcurves. Then

$$r + R = DF = L = \text{axes space} \qquad [1]$$

and because of rolling

$$r\,d\theta = R\,dw \qquad [2]$$

From Equation [2] we obtain

$$\frac{dw}{d\theta} = \frac{r}{R} \qquad [3]$$

However

$$\frac{dw}{d\theta} = \frac{\text{speed of follower}}{\text{speed of driver}} = \text{speedgraf ordinate}$$

Therefore, we obtain Equation [1] ($r + R = L$), and

$$r/R = \text{speedgraf ordinate} = S \qquad [4]$$

Combining Equations [1] and [4], we obtain

$$r = \frac{LS}{1 + S} \qquad [5]$$

and

$$R = \frac{L}{1 + S} \qquad [6]$$

As *L* is the axes space for the two gears it is a known constant. And as the speedgraf is known, *S* is known for every value of θ or for every driver displacement. We can therefore find as many different values of *S* as may be required. In a drawing, the angles θ are usually taken say every 5 deg. In computation for machine work, θ is taken for every 2 deg for gears say up to 10-in. axes space and for every 1 deg for larger gears.

The computation of *S* for every θ, of course, varies in the various segments of the speedgraf. In *AB* and *GH* the *S* is 0.0. In *DE*, *S* is constant. In the oblique *BC* it is equal to $(CC_0/B_0C_0) \times$ abscissa, measured from B_0. Similarly in the oblique *FG* it is equal to $(F_0F/F_0G_0) \times$ abscissa, measured back from G_0. In the parabolas *CD* and *EF* we use a computation as in a body falling under gravity.

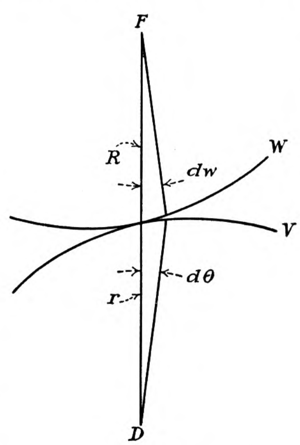

Fig. 8

We thus finally have the *S* value for every θ value and make a table for every θ, say from degree to degree.

Driver Rollcurve. We can now substitute in Equation [5] and obtain the value of the rollcurve ray for every θ. This gives the driver rollcurve mathematically accurate. The angle θ varies by equal angular steps.

Follower Rollcurve. By substituting in Equation [6] we obtain the set of follower rays corresponding to the follower sites for the driver places. The driver steps are equal. The follower steps usually are not equal and must be computed. These steps, of course, correspond to the corresponding area obtained by the speedgraf. We see that in the speedgraf the follower steps *AB* and *GH* are equal to zero. In *DE* they are constant. In *BC* and *FG* the areas are easily obtained as they are the areas of trapeziums. In CD and EF they are the areas by parabolic arcs, and can be computed say by Simpson's formula.

Thus, finally for every driver angle θ, there is found a corresponding angle w for the follower. This combined with R of Equation [6] gives the polar coordinates of the follower rollcurve.

Production of the Driver and Follower Blanks

Patterns and rough castings can be made from drawings made from the values previously computed herein. When it comes to accurate machining the shop facilities must be taken into account. The most direct method would be to use a turntable capable of being rotated and set to 0.001 deg. The ray length can then be measured to 0.0001 in. and a hole with a definite radius be bored. This would give a blank with a series of holes whose centers are exactly on the rollcurve.

In the establishment with which the author is associated there is no machine kept up to do the accurate boring as just outlined. There is, however, a very accurate jig boring machine with slides at right angles and it is kept in fine condition. Therefore, the polar coordinates previously obtained for the driver and follower rollcurves are now converted into rectangular coordinates. These are used on the jig borer and the rollcurve blanks are bored with centers exactly on the rollcurves.

Thus, by one machine and polar coordinates, or by another machine and rectangular coordinates, a first pair of blanks are obtained. The boring is carefully done and then the ridges between borings are filed off. A careful workman leaves very faint marks showing the bottoms of the original borings. This filed master will be called master cam No. 1. It is smaller than the rollcurve by the radius of the bore, or master cam No. 1 = rollcurve — bore radius of master cam No. 1.

The master cam No. 1 is now mounted on a cam reproducing machine. Of these, there is an excellent one in the shop. It really reproduces a cam. From master cam No. 1, by means of a proper sized roller and cutter, there is now produced master cam No. 2 whose periphery coincides with the rollcurve. Similarly by another operation on the same machine, there is produced a gear blank for the actual rollcurve gear. It has the addendum of the teeth added thereto.

For another set of gears, only a new gear blank has to be machined. Master cams Nos. 1 and 2 serve for all sets.

A similar procedure is now gone through for the follower rollcurve, and its master cam No. 1, and its rollcurve or master cam No. 2 are produced. From these as many blanks as desired with the teeth addenda added are milled out.

Computation of Arc Length

At first, attempts were made to approximate the rollcurves by circle arcs and to cut the teeth as circular segments. Here great difficulties were encountered. The approximations were made on the drawing board and it was immediately found that to obtain any acceptable result, the drawings had to be larger than natural. And there always was danger. It was very easy to go outside of the rollcurve and obtain a jam, or to go too far inside of the rollcurve and obtain undesired backlash. Finally, the author, discouraged with his ill success on the drawing board to obtain usable circle approximations, attacked the problem mathematically, and evolved the method now used.

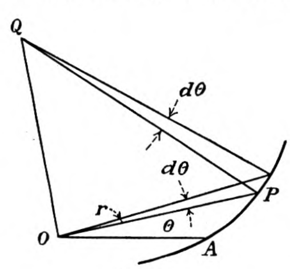

Fig. 9

In Fig. 9, let P be a point of a curve (rollcurve). Let $OP = r$, $AOP = \theta$. Let OQ be perpendicular to the ray OP, and PQ be a normal to the curve at P. Then PQ will be called the norm, and OQ the rayper(pendicular). In this place we call OQ the rayper, as being a short, expressive term.

We know from differential calculus that when θ is measured in radians (not degrees), dr and $rd\theta$ represent the ray increments along its length and transversely, respectively. This results immediately in the equation $dr/d\theta = OQ$. Moreover, the length of the arc, say AP is obtained thus

$$\text{arc } AP = \int_0^\theta \sqrt{(OP)^2 + (OQ)^2}\, d\theta = \int_0^\theta PQ d\theta$$

As the rollcurve AP was obtained from the speedgraf, and as the speedgraf was composed of horizontals, obliques, and parabolas, it seemed hopeful to obtain the polar equations of AP and to integrate for the length AP. There was no difficulty in obtaining the polar equations. The integration for the length of arc was something else. The horizontal segments gave simple integrals, the obliques gave elliptic integrals, and the fillet parabolas gave hyperelliptic integrals.

The author devoted much time to the study of the computation of elliptic and hyperelliptic integrals In this connection the best book he knows is "Simplification of Integrals," by Söderblom.[3] It is the best but is not simple enough. Therefore, the author was driven to the standard methods of approximation and used the well-known formulas, namely the Simpson $^1/_3$, the Simpson $^3/_8$, and the Weddle. These are given in various books.[4] Using these he was able to obtain the approximate values of the arcs. By computing first with the Simpson formula and then with Weddle's the author was able to make a check on the accuracy of the results. Making the original computations on a ten-place computing machine, his Simpson and Weddle results agreed to seven significant figures. These he assumed to be accurate. Moreover, they proved so in practice.

In this manner he obtained the length of the rollcurve arc of the driver from one step to another, say from degree to degree. As the lengths of the rollcurve arcs are the same for both driver and follower, he thus obtained also the length of the follower arc for degree to degree of the driver (not the follower). A table of the driver degrees and rollcurve arc lengths was then made.

The next was to select some satisfactory diametral pitch for the teeth of the rollcurve gears. This, of course, had already been done long ago when the proper addenda were added to the rollcurves to make the gear blanks. In practice the little gears in Fig. 2 had a diametral pitch of 16. For the bigger gears, there were used 8 pitch, 6 pitch, and 4 pitch, $14^1/_2$-deg involutes.

When the diametral pitch P was known, the pitch arc was known as $\frac{\pi}{P}$ and could be used as a measure to determine the number of pitch arcs.

Look again at Fig. 7. It shows the basic gear space on the driver. The start was therefore given, and all the other pitch points could be located therefrom.

Gage Points

Not all the pitch points were necessary. Only a few were required and the rest were derived from them. These were called the gage points and they had to be computed. These gage points almost never came at an exact step of the driver angle so did not fit in the table of arc lengths that had been computed. However, as the exact location of the gage points was required, the exact driver angle was computed by inverse interpolation.

[3] "Simplification d'Intégrales," by Axel Söderblom, Göteborg, Wald Zachrissons, Boktryckeri, A.B., 1909.

[4] "Numerical Mathematical Analysis," by J. B. Scarborough, Johns Hopkins Press, Baltimore, Md., 1931.

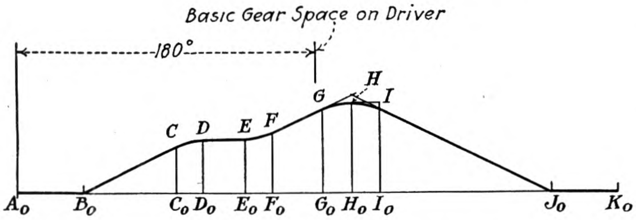

FIG. 10 DESIGNER'S SPEEDGRAF FOR A MORE COMPLICATED ROTARY GRIPPER DRIVE

A_0B_0 =	40.00000000 deg	A_0A =	0
B_0C_0 =	56.00000000 deg	B_0B =	0
C_0D_0 =	16.00000000 deg	C_0C =	1.20312500
D_0E_0 =	26.00000000 deg	D_0D =	1.37500000
E_0F_0 =	16.00000000 deg	E_0E =	1.37500000
F_0G_0 =	30.32939447 deg	F_0F =	1.54687500
G_0H_0 =	16.67060553 deg	G_0G =	2.19848308
H_0I_0 =	16.67060553 deg	H_0H =	2.37756185
I_0J_0 =	102.32939447 deg	I_0I =	2.19848308
J_0K_0 =	40.00000000 deg	J_0J =	0
A_0K_0 =	360.00000000 deg	K_0K =	0

Axes distance = 14.2500 in.
Arm for roller
Arms angle
Diametral pitch = 8
Gear cam equal steps = 1 deg
Roller cam equal steps = 1 deg
Roller diameter

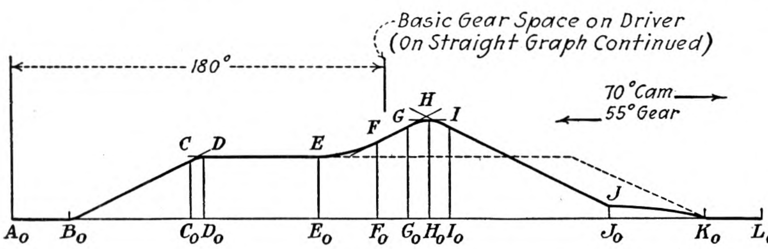

FIG. 11 SPEEDGRAF FOR A 15-IN. TAPE SPEED OF A ROTARY GRIPPER DRIVE

	Driver angle, deg	Follower angle, deg		Follower velocity
A_0B_0 =	27.6522	0	A_0A =	0
B_0C_0 =	60.0000	40.872131	B_0B =	0
C_0D_0 =	6.0000	8.898361	C_0C =	1.4327868
D_0E_0 =	55.0858	83.080223	D_0D =	1.5081967
E_0F_0 =	28.2843	46.009840	E_0E =	1.5081967
F_0G_0 =	14.3077	29.237877	F_0F =	1.8636821
G_0H_0 =	10.0000	23.071194	G_0G =	2.2233307
H_0I_0 =	10.0000	23.071194	H_0H =	2.3490138
I_0J_0 =	76.0178	96.384139	I_0I =	2.2233307
J_0K_0 =	45.0000	9.375000	K_0K =	0
K_0L_0 =	27.6522	0	L_0L =	0
A_0L_0 =	360.0000	359.999959		

Pearson Quadratic

After spending a considerable time experimenting with various interpolation formulas, the author selected the Pearson quadratic as being the most suitable for this work. It can be found in "Tracts for Computers," No. 2, University of London, 1920. It is the simplest he knows of, for both direct and inverse interpolation. Moreover, both Simpson and Weddle formulas can be derived from it.

By means of the quadratic, all interpolations were made, and finally the ray lengths and ray angles were converted into rectangular coordinates by means of trigonometric tables. The values were then entered on the detail drawings.

Cutting of Teeth

It has been stated that a Fellows gear shaper cutter was used to cut the teeth on the gears. This was done by making a special machine in which the master cam No. 2, already described, was used. As a roller for it, rolled around the master, the cutter shaped out the teeth of the gear blank.

It was soon found that sometimes the roller had to be increased or decreased in diameter. Here the gage points came in. The cutting was started at the start gage point. If the finish gage point were hit accurately, it would be assumed that the intermediate points were correct. In practice it always worked well enough.

This first gear cut was seldom used in the drive. It was installed in the gear-cutting machine. In it then rolled a toothed circular gear, nonrotatable with the Fellows cutter. All subsequent gears were thus "fellowsed" without the possibility of slipping.

Refer now to Fig. 5. Note that the gears are both cut off. This was usually done at a place where the ray-norm-angle (of both the driver and follower) was 45 deg. If the angle were larger, the gears would unmesh. The complete drive for the speedgraf was thus unfinished. The drive for the gap was made by two cams on the driver and two rollers on the follower. These are seen in several of the gear illustrations. However, the making of roller cams for a given drive is nothing new and so it is not gone into further.

As it has been stated before that by means of the rollcurve gears almost any desired motion of the follower may be obtained for example, the author adds Fig. 10 showing a more complicated speedgraf. In making the rollcurve gears therefrom the work was slightly more extensive but no new principles were encountered.

Also, finally, the author shows an even more complicated speedgraf Fig. 11. Here the cusp J was in the cam part, not in the rollcurve gear part. Moreover, the designer checked both the driver and follower angles himself.

The foregoing gives a general description of the subject. In practice it was necessary to systematize it, and to reduce it to a method that could be used say by an intelligent high-school graduate. The details of this, however, would require an article much larger than the space taken herein, and it has been thought that sufficient has been given for the present.

Discussion

J. W. HUCKERT.[5] Spur gears having pitch curves other than circles have been in demand for a long time. One obstacle is the mathematics. It is too complicated for practical use. The rectification of the noncircular curves, as the author points out, involves elliptic and hyperelliptic integrals, and much labor is required to evaluate them either by the methods of the calculus or by the methods of approximation. If the methods of the author are simple enough so that he can compute the chordal thicknesses of all the teeth on a rollcurve gear, it may be enlightening to compare the theoretical against the measured thicknesses as a check on the spacing of the teeth.

In order to have accurate spacing of the teeth, the angular-speed ratios proper between the tool and the blank must be maintained throughout the generation of the gear. It is doubtful whether the author has met this requirement so that gears may be generated which will be suitable for all classes of machinery. In the ordinary setup of the Fellow's gear-shaper, the tool and the blank are positively rotated with respect to each other by means of a gear train. The author replaces this gear train with a friction drive during the cutting of the first gear. If slippage occurs, the spacing of the teeth on the first gear cannot be accurate. Since the first gear is employed in the drive between the tool and the blank during the cutting of subsequent gears, its

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inaccuracies will be reflected in all the others. Maybe accuracy in spacing is not necessary on gears for printing machinery. Neither is it so important on gears for agricultural, textile, and wrapping machinery. However, accuracy of spacing is important on gears for high-speed machinery and for machine tools, where not only the total angular displacements but also the instantaneous angular speeds must be preserved.

A. L. Ruiz.[6] This paper presents the practical aspects of a topic the writer has examined at various times in the past from a theoretical point of view. The investigations were never carried through to the stage of constructing noncircular gears, principally because of lack of an immediate application. It was with interest therefore, that the writer read the author's exposition of his method of manufacturing rollcurve gears. It is requested that the author explain further certain points and an alternative method of using these gears under certain circumstances is proposed.

With regard to cutting rollcurve gears on the Fellow's gear-shaping machine, it is rather clear that the cutter is moved in and out with respect to the blank by means of a roller bearing on a master cam. It is not so clear to the writer how the proper rotations of the cutter and the blank are obtained. In cutting the first of a particular type rollcurve gear, it appears that the cutter alone is rotated, and this, as it cuts into the blank, causes it to rotate. When one successful gear has been produced, it is used in the machine with a proper-size gear to cause succeeding blanks to rotate at a definite variable rate with respect to the cutter. Is this interpretation of the process correct?

Mention was made in the paper of the calculation of arc lengths of the rolling curves for the determination of gage points. Is this calculation extended to determine the size of the rolling curves so that an integral number of teeth is obtained in a complete circumference of each gear?

The author is apparently interested primarily in using the rollcurve gears to obtain a variable rotational velocity from a constant rotational velocity. With what degree of accuracy do the gears produce the velocity for which they were designed? For other applications one might be interested in obtaining a variable position of the follower shaft as a function of the position of the driving shaft. To what accuracy do rollcurve gears perform this service? The problem of backlash is of prime importance in any system designed to reproduce positions. This was touched on during the verbal discussion of the paper at the Annual Meeting. At that time the author stated that his rollcurve gears had backlash which increased as the ray-norm angle increased, although no magnitudes were given. The thought immediately arises as to whether methods for reducing the backlash have been tried, such as, decreasing the center distances of the gears, or using slightly enlarged blanks, or cutting thicker teeth.

The paper states that for low velocity ratios or for zero velocity of the follower gear, roller cams were used; the curved gears were designed to unmesh at a proper place, at which point the roller cam assumed the duty of transmitting the variable rotary motion. It would appear that the design and manufacturing requirements for constructing cams and gears so as to give a smooth transition with a minimum of lost motion would be rather fussy. The writer suggests the following scheme as a possible antidote for these difficulties.

Assume the designer wishes to construct a pair of rollcurve gears for a speedgraf some of whose ordinates are zero or negative. Add a constant to each ordinate so chosen that the entire speedgraf becomes positive. All ordinates may then be scaled down in proportion so that the area under the graph will be 360 deg. If the constants have been chosen properly, a pair of rollcurve gears may be constructed for this resulting speedgraf. For this pair of gears, as the driver is rotated at a constant velocity, the follower never reverses, but always rotates in a direction opposite to that of the driver, with a velocity which varies between predetermined maximum and minimum values. Then, connect the driving gear and the following gear together differentially with some form of planetary gearing, so that a portion of the driver velocity is subtracted from the follower velocity. If the gear ratios are chosen properly, the output of the differential will rotate at a velocity proportional to the ordinates of the original speedgraf. (Adding a constant to every ordinate of the speedgraf means adding a constant to the velocity of the follower gear at each point. This constant velocity is subtracted out by the planetary gearing.) The advantage of this method of construction is that the roller cam could be eliminated. The motion would be transmitted entirely by rollcurve gears and straight gearing, with a probable decrease in lost motion and an increase in smoothness of operation.

[6] Aeronautics and Marine Engineering Department, General Electric Company, Schenectady, N. Y. Jun. A.S.M.E.

Author's Closure

The author wishes to clear up some questions asked in the oral discussion of the paper at the time of its presentation. One was in reference to the backlash where the ray-norm angle is increased. The author answered that the backlash was increased. Somehow this answer led some to the idea that thereby a good deal of backlash was introduced. This is a false conclusion. If the gear is cut with a backlash of 0.0000 in. where the ray-norm angle is 0, and with similar accuracy at other places, then the backlash would be 0.0000 in. everywhere even at 45 deg. If the backlash were, say, 0.001 in. where the ray-norm angle is 0, then when the angle is 45 deg (at the limit) the backlash would be only $\sqrt{2} \times 0.001$ or 0.001414 in. The author's meticulous truthfulness and accuracy seem to have misled the listeners. For all practical purposes there is no backlash at all, anywhere. In ordinary spur gears, we work with backlash of, say, from 0.000 to 0.010 in. The rollcurve gears have been made equally tight.

A second point raised at the meeting was that in the rollcurve gear, the line of pressure of the gear tooth changes all the time and, therefore, the theory cannot be right. However, in cycloidal gears, the line of action is not straight and nevertheless the action is correct. The same is true with rollcurve gears.

The third point raised at the meeting was that the use of a friction drive at one point, obviously made the process inaccurate so that it could not be exact. Here of course the question arises as to what is meant by "exact." Must gears be correct in every dimension to an accuracy of 0.000001 in. before they are called *exact?* If so, then these rollcurve gears are not exact. But there never has been any exact gear made by anybody at any time in this world so far as the author knows. In commercial spur gears in the chord over, say, five teeth, the variations will amount to say 0.005 in., and the rollcurve gears are equally accurate.

The author does not know of any better test than by the accurate fitting of the plug in the predetermined space of the rollcurve gear, giving a location to the plug accurate in both abscissa and ordinate. It is actually correct to about 0.0005 in. and that is close enough for even the fine work such as is required for printing presses.

Professor Huckert states "Maybe accuracy of spacing is not necessary . . . . on gears for printing machinery." Actually, printing presses require an accuracy of spacing that is beyond the commercial output of even the best of machine gear cutters. The teeth have to be corrected as by the Maag gear-tooth grinder.

The author does not understand the meaning of ". . . where not only the total angular displacements but also the instantaneous angular speeds must be preserved." The author always assumed that if the angular displacements are always right, the instantaneous angular speeds must always be right. In any case in so far as the rollcurve gears are concerned, both the instantaneous angular displacements and also the instantaneous angular speeds are always preserved.

That the mathematics is "too complicated for practical use" raises quite a discussion. It would be spendid to find simpler mathematics for the solution of the problem. The author does not know any simpler. *The fact remains, however, that it is being used by the Miehle Company to make gears on their lines of printing presses. If that is not practical use, what is?*

Mr. Ruiz seems to have concluded that the rollcurve gears are cut on a Fellows gear shaper. They are not. The author nowhere said so. He did say that a Fellows gear-shaper cutter was used. The gears are cut on a special machine designed for the purpose, which it would take far too long to attempt to describe.

Complete gears, with an integral number of teeth, have been made, as in Fig. 2, and are being made regularly. There are now cuspless, continuous rollcurve gears, driving each other as desired. There is no difficulty.

It is true that the author described the driver rollcurve gear as rotating at a uniform rate, and the follower as having a predetermined variable rate. But this is simply a mathematical convenience. After the gears, have been cut and meshed, then obviously the driver may be moved and at any rate, constant or variable.

Again Mr. Ruiz also raises the question of accuracy. The gears are machined to 0.001 or 0.002 in. on the rollcurve arc and they perform their function to this degree of accuracy. No special means for decreasing the backlash is required since, when desired, the gears are cut to run metal to metal without backlash.

Mr. Ruiz intimates that it is difficult to make the node cams join the gears smoothly. "It must be a fussy job." It is a commercial job, as the line of machinery marketed by the firm shows. It gives a fine job with a minimum of backlash.

Mr. Ruiz inquires whether by the use of an auxiliary planetary gear, it would not be possible to produce a dwell and even a negative rotation. Certainly it would be possible, but the author abandoned that idea because in his opinion it would introduce not a minimum of backlash but an increased backlash. Moreover the machine, Fig. 3, produced both positive and negative rotations of the follower. To accomplish this result an intermediate (but not planetary) gear was used, and is being used today.

Moreover, when the gears need to be on a dwell in printing-press work, there should be a dwell and not a constant quiver.

Finally, the author does not doubt that improvements can be made in the mathematics and in the method of cutting. He himself has shown that it is both possible and practical to produce rollcurve gears. He assumes this is but the beginning and sees no reason why rollcurve gears should not become quite common.

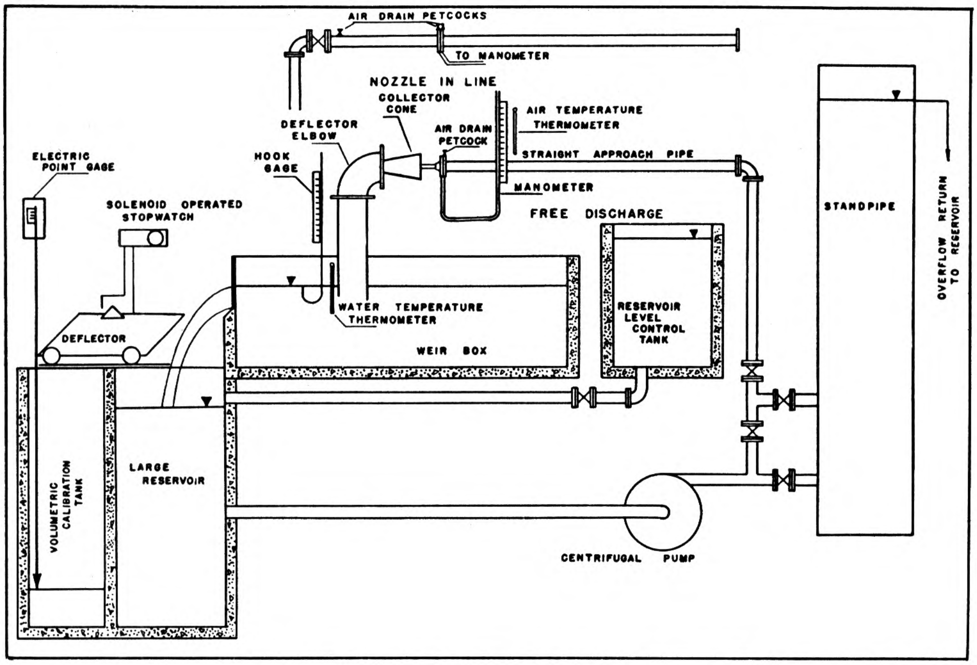

Fig. 1 Arrangement of Test Equipment at the University of California for the Calibration of Nozzles Placed in and at the End of a Pipe Line

Nozzle Coefficients for Free and Submerged Discharge

By R. G. FOLSOM,[1] BERKELEY, CALIF.

This paper presents in brief form a summary of investigations made at the hydraulic laboratory of the University of California, in cooperation with the A.S.M.E. Special Research Committee on Fluid Meters, to determine nozzle coefficients for free and submerged discharge. A description of test facilities and procedure is included. The results given concern a series of A.S.M.E. long-radius nozzles for 3-in., 4-in., and 8-in. pipes tested with pipes of different wall roughness, various upstream piping arrangements, and different pressure-tap locations. Some data on I.S.A. nozzles which have been manufactured at the University shops are added.

THE INVESTIGATION reported in this paper is one of many conducted at different laboratories throughout the country with the support of the A.S.M.E. Special Research Committee on Fluid Meters to study the characteristics of the A.S.M.E. long-radius nozzle. Earlier papers have dealt with the general program as well as some particular results (1, 2, 3).[2]

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[2] Numbers in parentheses refer to the Bibliography.

Contributed by the A.S.M.E. Special Research Committee on Fluid Meters and presented at the Annual Meeting of The American Society of Mechanical Engineers, held in New York, N. Y., December 5–9, 1938.

Discussion of this paper was closed January 10, 1939, and is published herewith directly following the paper.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The work at the University of California involves calibration of the nozzles placed in a pipe line in the normal manner, and also placed at the end of the pipe line. The latter condition of free discharge is of importance in various testing arrangements, particularly in tests of irrigation and dredge pumps. A previous paper (4) presented results of tests with orifices for free discharge.

During the investigations, it was found that if the nozzles were left in the pipe line for a few days in contact with water, material forming a sand-like surface was deposited on the nozzles. A similar phenomenon has been observed at the National Bureau of Standards (3). This deposit has an appreciable effect on the coefficient. Unless otherwise noted, all tests considered in this paper refer to nozzles with clean smooth surfaces.

Equipment and Procedure

Fig. 1 shows the arrangement of the equipment in the Hydraulic Laboratory at the University of California for calibration of nozzles placed in and at the end of a pipe line. Water was drawn from the large reservoir by the centrifugal pump and discharged to the standpipe or directly to the nozzle as desired. An additional reservoir was provided to maintain constant suction conditions on the centrifugal pump when it discharged directly to the nozzle, thus keeping the discharge rate constant during the test.

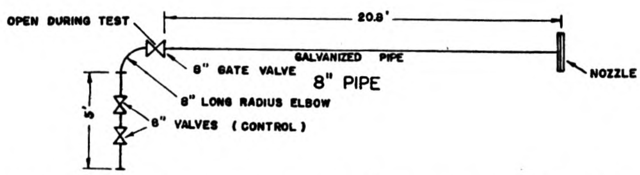

FIG. 2 ACTUAL ARRANGEMENT AND DIMENSIONS FOR TESTS WITH 8-IN. GALVANIZED PIPE WITH FREE DISCHARGE

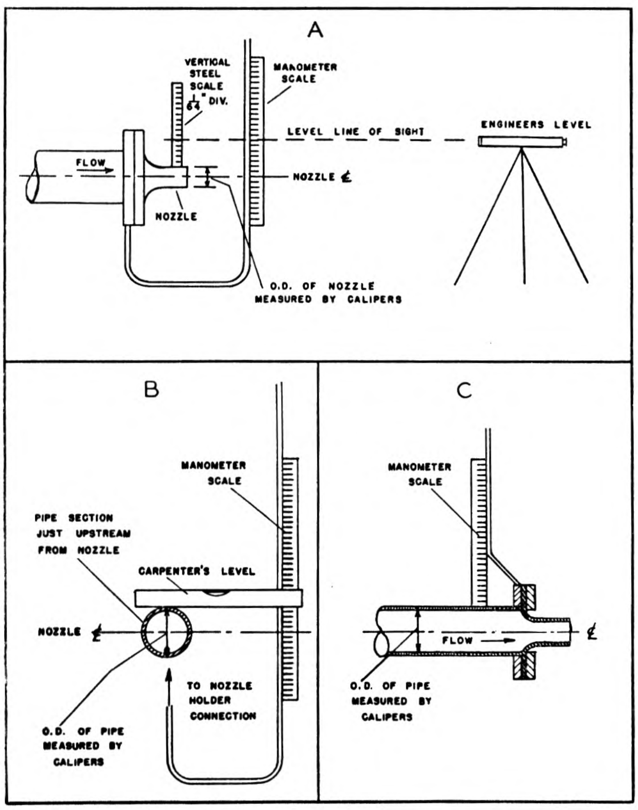

FIG. 3 VARIOUS METHODS FOR DETERMINING THE ZERO READING OF MANOMETERS

The dimensions and actual piping arrangement for a typical test setup are shown in Fig. 2.

Nozzles. The long-radius nozzles supplied to the laboratory were manufactured from brass or bronze and possessed smooth surfaces except as indicated in the following:

Nozzle No. 8.300.2 was built up by brazing parts together and then machining the meter. Since the material was not homogeneous from the standpoint of hardness, the tool tended to chatter and left a series of waves on portions of the nozzle converging section.

Nozzle No. 8.375.1 was similar in construction to that of nozzle No. 8.300.2. Blowholes were evident at the junction of the parts which also left a prominent line around the upstream face.

Nozzle No. 8.562.1 was made of steel.

Nozzle No. 8.506.1 had indentation areas in the surface, which were due to insufficient metal in the casting.

Quantity Measurements. The rate of discharge was determined from the volume of water collected in a measured period of time.

The gravimetrically calibrated volumetric tank had a capacity of about 400 cu ft and a cross-sectional area of about 46 sq ft. The rise of water level was measured with an electric point gage reading to the nearest 0.001 ft. The water-collection period was determined from a solenoid-operated stop watch with contacts for the solenoid placed on the hand-operated deflector. The stop watch was checked daily against a clock known to keep correct time within a few seconds per month.

The hook gage was read at the start and finish of each run to determine the storage or depletion of water in the weir box. The weir was not used as a flow-measuring device.

Head Measurements. All nozzles were mounted in holding rings having centering devices and with corner taps which corresponded to I.S.A. standards (5, 6). Additional pipe taps were placed at various distances upstream for the free-discharge measurements. For the nozzle-in-line position, corner taps, and pipe taps, one pipe diameter upstream and one-half pipe diameter downstream, were used.

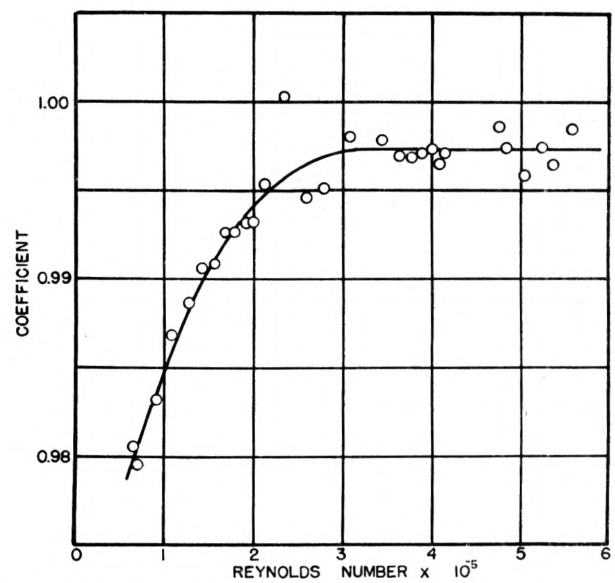

FIG. 4 TYPICAL NOZZLE-CALIBRATION CURVE PLOTTED FROM TEST DATA ON NOZZLE NO. 57195*R* WITH GALVANIZED PIPE AND FREE DISCHARGE

Vertical water columns, slant and vertical mercury-filled U-tubes were used for manometers. The scales were graduated to 0.01 ft and all readings were estimated to the nearest 0.001 ft. The inside diameters of the manometer tubes were of sufficient size to make the effects of surface tension negligible (7).

With free discharge, it is necessary to measure the elevation of the manometer scale with respect to the center line of the nozzle. Fig. 3 illustrates some common methods used for this determination. The method shown in Fig. 3*A* was employed for the tests reported in this paper.

In many cases, the room or air temperature was appreciably different from the water temperature, the discrepancy reaching as high as 30 F. For these tests, the manometer fluid was assumed to be at room temperature. Corrections were applied to express the head in terms of the fluid flowing through the nozzle.

EXPERIMENTAL RESULTS

Discharge Equation. As indicated elsewhere (4), the form of the discharge equation is largely a matter of convenience. All coefficients reported in this paper were computed from

$$Q = CA\sqrt{(2gH)} \quad \text{[1]}$$

TABLE 1 EXPERIMENTAL RESULTS WITH CORNER TAPS

			Free Discharge								In-line								
			Galvanized pipe				Black pipe (service rough)				Galvanized pipe				Black pipe (service rough)				
Nozzle no.	Nozzle diameter in.	Diam ratio, D_2/D_1	Straight pipe, no. diameters	C	Avg deviation, per cent	Min RN $\times 10^{-5}$	Straight pipe, no. diameters	C	Avg deviation, per cent	Min RN $\times 10^{-5}$	Straight pipe, no. diameters	C	Avg deviation, per cent	Min RN $\times 10^{-5}$	Straight pipe, no. diameters	C	Avg deviation, per cent	Min RN $\times 10^{-5}$	Coefficient ratio[b]
57195R	1.2240	0.400	81	0.997	0.10	3.0	..	...	..	...	..	...	..	...	..	...	..	...	...
57299	2.4476	0.800	81	1.165	0.38	2.0	..	...	..	...	..	...	..	...	..	...	..	...	...
4.150.2	1.4374	0.356	60	0.991	0.14	2.9	58	0.989	0.13	1.8	..	...	..	...	..	...	..	...	...
4.200.4	2.000	0.496	60	1.004	0.09	1.7	58	1.007	0.07	1.5	60	0.991	0.19	3.2	..	...	..	...	1.000
4.250.1	2.5007	0.619	60	1.044	0.12	3.0	59	1.045	0.10	2.5	60	1.005	0.11	1.5	48	1.009	0.08	1.8	0.999
4.275.1	2.7502	0.681	60	1.068	0.23	3.0	59	1.076	0.18	2.5	60	1.045[a]	...	5.5	..	...	..	...	0.999
4.300.1	3.001	0.743	60	1.096	0.10	2.3	58	1.116	0.13	1.6	60	1.068	0.16	2.0	..	...	..	...	1.000
4.350.1	3.5001	0.867	60	1.306	0.37	1.9	59	1.314	0.22	4.0	60	1.102	0.08	2.2	48	1.118	0.19	2.5	0.995
8.300.1	3.0012	0.371	30	0.996	0.14	1.8	20	0.993	0.13	2.0	30	0.995	0.22	2.3	..	...	..	...	1.001
8.300.2	3.1226	0.386	30	0.996	0.15	3.0	20	0.994	0.08	3.0	30	0.995	0.15	3.0	..	...	..	...	1.001
8.375.1	3.7497	0.464	30	1.005	0.09	2.3	20	1.007	0.07	2.3	30	1.003	0.16	3.0	..	...	..	...	1.002
8.450.1	4.4981	0.557	30	1.018	0.17	3.0	20	1.024	0.22	2.3	30	1.019	0.13	2.3	..	...	..	...	0.999
8.506.1	5.0585	0.626	31	1.037	0.18	4.1	20	1.040	0.20	2.8	30	1.037	0.30	4.7	..	...	..	...	1.000
8.562.1	5.6180	0.696	31	1.068	0.35	3.2	20	1.076	0.30	3.4	30	1.071	0.29	3.6	..	...	..	...	0.997
8.615.1	6.150	0.761	31	1.123	0.31	6.1	20	1.134	0.12	5.0	30	1.120	0.18	6.8	..	...	..	...	1.003
8.665.1	6.648	0.823	31	1.187	0.24	7.0	20	1.214	0.46	6.7	30	1.180	0.18	6.5	..	...	..	...	1.005
8.715.1	7.151	0.886	31	1.321	0.60	5.6	20	1.371	0.47	6.7	30	1.300	0.42	4.3	..	...	..	...	1.016

[a] The coefficient did not reach a well-defined constant magnitude. The value given is an approximate one toward which the coefficients approach.
[b] Ratio of coefficients for nozzle at free discharge to the coefficient in-line for galvanized pipes.
RN = Reynolds' number.

where Q = discharge rate, cfs; C = a dimensionless coefficient; A = area of nozzle throat, sq ft; g = weight per unit mass = 32.15 ft per sec per sec; and H = differential head across the nozzle (in-line) or head at the center of the nozzle (free discharge) expressed in feet of water corresponding to pipe-line conditions.

The coefficient C is primarily a function of the diameter ratio, upstream-velocity distribution, and the pressure-tap location and

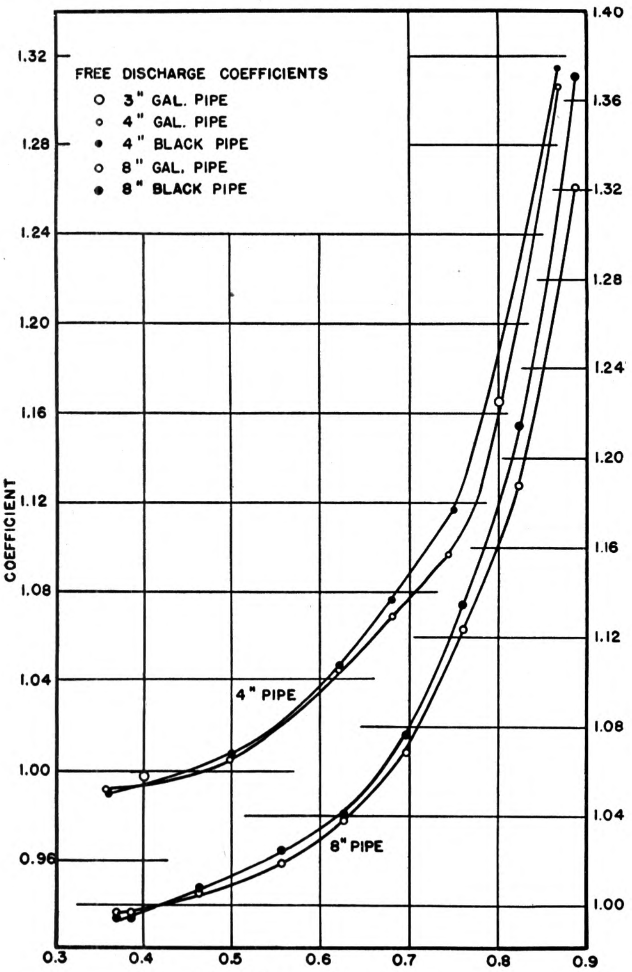

FIG. 5 FREE-DISCHARGE COEFFICIENTS WITH CORNER TAPS FOR NOZZLES IN 3-IN., 4-IN., AND 8-IN. GALVANIZED AND BLACK PIPES

type. As the upstream-velocity distribution is often difficult to measure, the factors which influence the distribution are usually indicated as the variables. These are the Reynolds number, roughness and length of approach section, and piping arrangement upstream from the straight approach section. Other factors also influence the coefficient, e.g., roughness of nozzle, submerged or free discharge, and eccentricity of the nozzle with respect to the pipe. The linear dimension in the Reynolds number was selected as the throat diameter of the nozzles.

Free Discharge With Corner Taps. Fig. 4 shows a typical nozzle-calibration curve, that is, the coefficient rises at low Reynolds number until a critical value is reached. At larger Reynolds numbers the coefficient is constant. The critical value is indicated as the minimum Reynolds number for a constant coefficient, in this case about 3×10^5. Coefficients used in this report refer to values in the constant-coefficient region. Insufficient data were obtained to determine a minimum Reynolds-number curve throughout the diameter ratios investigated.

The summary of tests with new galvanized and service-rough[3] or new black pipe is presented in graphical form in Fig. 5. Specific values are tabulated in Table 1. The values for the largest diameter ratios are approximate since repeat tests did not check original values. For these diameter ratios ($D_2/D_1 > 0.82$, where D_1 is pipe and D_2 is the nozzle diameter), the coefficient is extremely sensitive to conditions of nozzle surface, eccentricity with the pipe, and pipe roughness.

In Line With Corner Taps. The usual coefficients for quantity rate meters refer to installations in the middle of a pipe line. Table 1 contains a column of coefficients obtained for this type of application. Previous treatments of free-discharge conditions have expressed the coefficients in terms of the coefficients for the in-line installation. Table 1 shows this comparison as obtained from this series of tests. The 4-in. nozzles show an average ratio slightly less than unity, while the average for the 8-in. nozzles is slightly greater than unity, the average of all runs being close to unity. Thus, according to this investigation, the coefficients for the in-line and free-discharge operation differ by a negligible amount.

Pressure-Tap Location. The type and position of pressure taps depend on many factors. The experience of some laboratories has indicated that a well-made pipe tap is less sensitive to extraneous influences than corner taps. On the other hand, experience has demonstrated the difficulties involved in produc-

[3] Translation from reference (6): "A service-rough pipe may be understood to be a cast-iron pipe which has rusted internally through long use, but which has no thick encrustations on the surface."

ing a well-made pipe-tap connection under field conditions. In view of the uncertainty regarding the optimum position, investigations were made to determine the relationships of heads measured at various points.

The method selected to present the results of upstream pressure investigations assumes that if the coefficient based on the corner tap were constant, then the coefficients based on other taps investigated would be constant also. Analysis of the original data indicates this procedure to be valid. The results are expressed in terms of a ratio of heads based on the head measured at the corner tap.

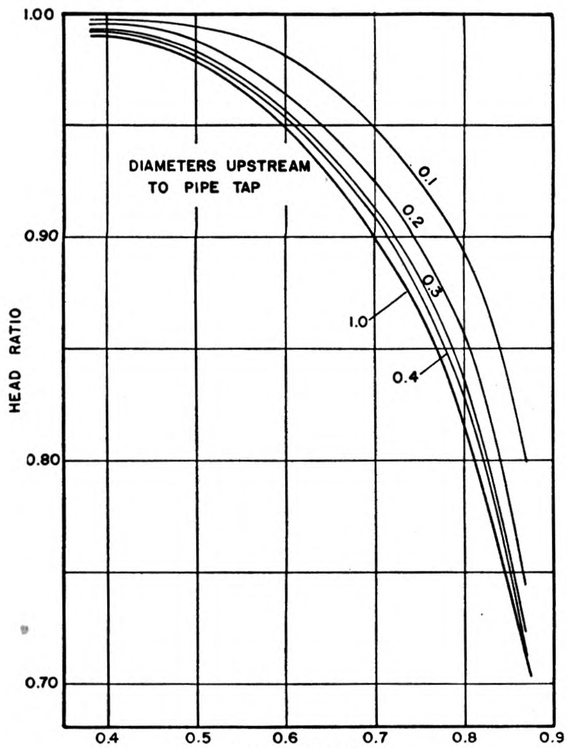

FIG. 6 RATIO OF HEADS AT UPSTREAM PIPE TAPS TO HEADS AT CORNER TAPS FOR FREE DISCHARGE

The results of head measurements in galvanized pipes at the corner tap, and taps about 0.1, 0.35, 0.5, 0.6, and 1.0 diameters upstream are shown in the faired curves of Fig. 6. The square root of the pressure ratio will give the change in coefficient at any upstream distance with respect to the coefficient based on the corner tap for corresponding diameter ratios. For example, at a diameter ratio of 0.62 and a distance of 1 diameter upstream the pressure ratio is 0.94. If the coefficient for the corner tap were 1.033, then the coefficient for one diameter upstream would be $1.033 \div (0.94)^{1/2}$ or 1.065.

The pressures measured at 1 diameter upstream from the upstream face of the nozzle and $1/2$ diameter downstream from the downstream side of the nozzle plate have been shown to be the most satisfactory (3). Several of the nozzles were tested in the line with these pipe taps. Also tests on free discharge with 1 diameter upstream pipe taps were made. The results for both series of investigations are given in Table 2 and Fig. 7.

TABLE 2 RATIO OF HEADS FOR PIPE TAPS AND CORNER TAPS

		Head Ratio			
		Free discharge[a]		In-line[b]	
Nozzle no.	Diameter ratio, D_2/D_1	Gal. pipe	Black pipe	Gal. pipe	Black pipe
57299	0.800	0.825	...	...	...
4.150.2	0.356	...	...	...	...
4.200.4	0.496	0.979	0.981	0.985	0.980
4.250.1	0.619	0.943	...	...	...
4.275.1	0.681	0.918	...	...	...
4.300.1	0.743	0.863	0.874	0.877	0.879
4.350.1	0.867	0.768	...	...	...
8.300.1	0.371	0.992	...	...	...
8.300.2	0.386	0.990	0.991	...	...
8.375.1	0.464	0.984	0.987	...	...
8.450.1	0.556	0.960	0.957	...	...
8.506.1	0.627	0.933	0.937	0.924	...
8.562.1	0.696	0.898	0.903	0.892	...
8.615.1	0.762	0.851	0.866	0.850	...
8.665.1	0.823	0.783	0.807	0.777	...
8.715.1	0.886	0.692	...	0.706	...

[a] Ratio of head at 1 diameter upstream to that at the corner tap.
[b] Ratio of differential heads measured at corner taps to that measured with taps 1 diameter upstream and $1/2$ diameter downstream.

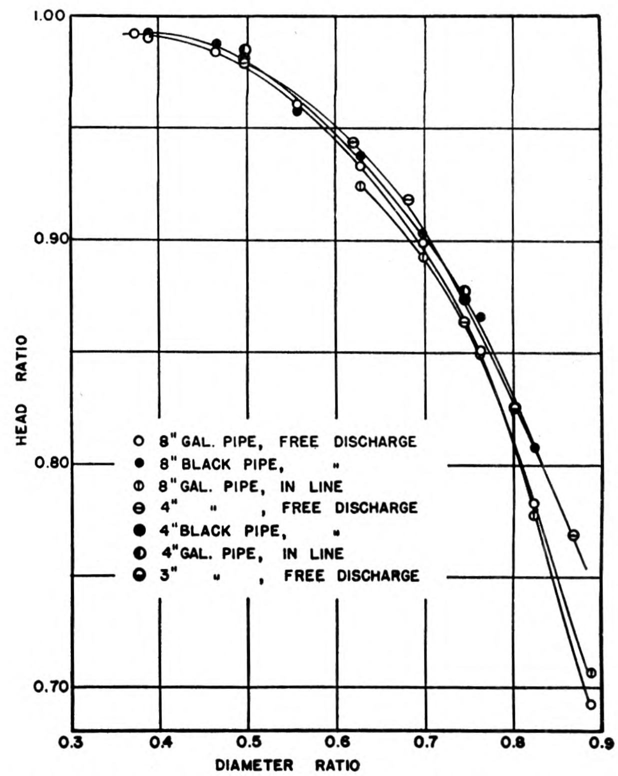

FIG. 7 RATIO OF HEADS AT A PRESSURE TAP 1 DIAMETER UPSTREAM TO THE HEAD AT THE CORNER TAP FOR FREE DISCHARGE: ALSO, THE RATIO OF HEADS AT 1 DIAMETER UPSTREAM AND $1/2$ DIAMETER DOWNSTREAM TO THE HEAD WITH CORNER TAPS FOR IN-LINE POSITION

Short Approach Lengths. In actual fluid measurements, it is frequently necessary to install a quantity rate meter with insufficient upstream straight approach pipe. Specifications for straightening vanes have been proposed (8), but the straight lengths of pipes specified are longer than desired. Data on some nozzles installed a short distance downstream from a single elbow were obtained to demonstrate the effects of distorted velocity distribution on the coefficients. All measurements were made with corner taps and black or service-rough standard steel pipe; the results are shown in Fig. 8. Throughout the diameter ratios studied, the coefficient reduces as the length of upstream pipe decreases, the reduction being greater the larger the diameter ratio.

A curve for nozzle coefficients with a highly corroded pipe is also shown in Fig. 8. As the pipe becomes rougher, the coefficient rises, the rise increasing with the diameter ratio.

I.S.A. Nozzles. Nozzles manufactured in the laboratory shops were calibrated in order to compare coefficients with standards specified in the "Regeln" (6). Bronze or brass castings were machined to required dimensions as indicated by a template and given a high polish. No measurements were made on the actual contour of the finished nozzle. Table 3 presents the test results and the comparison with standard values. The *a* value for the 3-in. nozzle was measured about 3 months prior to the *b* value, the pipe and holder in each case being the same. During the interval, the nozzle had been used but no deposit developed and the surface was merely polished for the *b* series of tests.

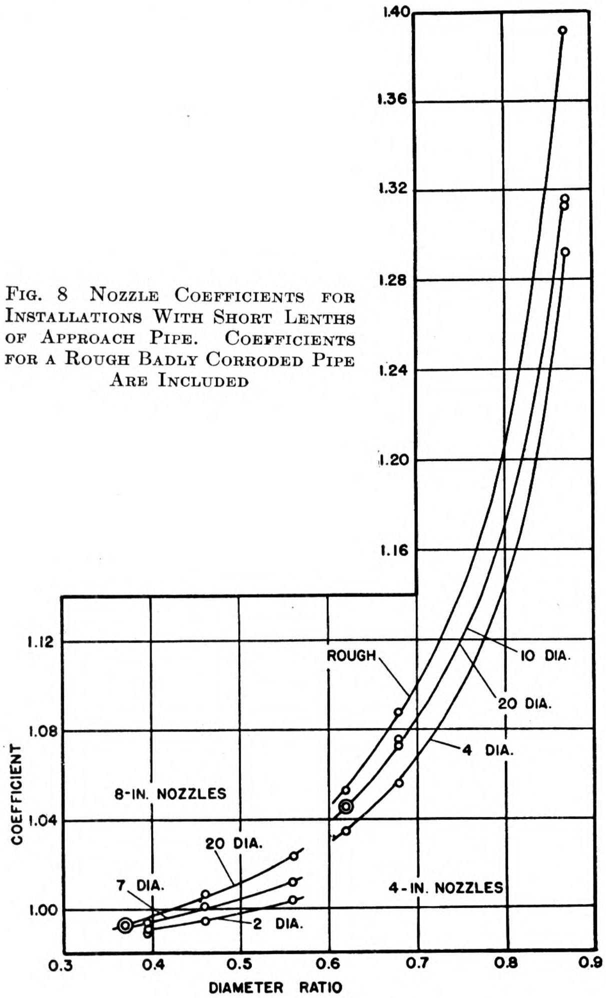

FIG. 8 NOZZLE COEFFICIENTS FOR INSTALLATIONS WITH SHORT LENTHS OF APPROACH PIPE. COEFFICIENTS FOR A ROUGH BADLY CORRODED PIPE ARE INCLUDED

Nozzle no.	D_2/D_1	Straight pipe, no. diameters	C
8.300.1	0.37	20	0.993
		7	0.993
8.300.2	0.39	20	0.994
		7	0.991
		2	0.989
8.375.1	0.46	20	1.007
		7	1.001
		2	0.995
8.450.1	0.56	20	1.024
		7	1.012
		2	1.004
4.250.1	0.62	20	1.045
		10	1.045[a]
		4	1.035[a]
4.275.1	0.68	20	1.076
		10	1.073
		4	1.056
4.350.1	0.87	20	1.314
		10	1.312
		4	1.292
Rough badly corroded pipe:			
4.250.1	0.62	40	1.053
4.275.1	0.68	40	1.088
4.350.1	0.87	40	1.392

[a] The coefficient did not reach a well-defined constant magnitude.

TABLE 3 I.S.A. NOZZLE WITH CORNER TAPS

Nozzle no.	Nozzle diameter, in.	Diameter ratio, D_2/D_1	Coefficients: Free discharge, gal pipe	Coefficients: In-line, gal pipe	Coefficients: C, "Regeln"[a]	Coefficients: C, corrected	Ratio of free-discharge coefficient to C	Ratio of free-discharge coeff to in-line coeff
3 in.	1.890	0.62	(a) 1.041 (b) 1.033	...	1.036	1.0380	...	...
HN-7	2.493	0.62	1.064[b]	...	1.036	1.0375	...	...
PN-1	2.400	0.60	1.024	1.025	1.030	1.0310	0.994	...
PN-2	4.445	0.56	1.007	1.010	1.018	1.0180	0.992	0.999
PN-3	1.253	0.61	...	1.029	1.033	1.0360	0.993	0.997

[a] Correction of coefficients for pipe roughness. See paragraph 28 and Fig. 23*a* of reference no. 6 in the Bibliography.
[b] This coefficient corresponds to a rough nozzle.

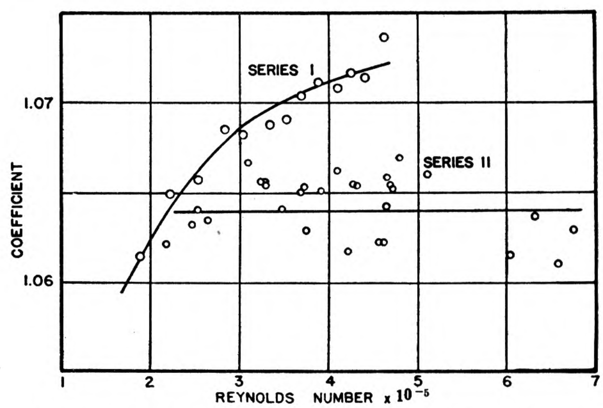

FIG. 9 CALIBRATION DATA ON NOZZLE NO. HN-7 FOR FREE DISCHARGE

(Series I tests are for a smooth nozzle surface, and series II tests apply when deposits were present on the nozzle surface.)

Fig. 9 shows the data for two tests on the HN-7 nozzle. Series I corresponds to new highly polished conditions. Series II is the calibration after the nozzle had been installed in a regular laboratory service line long enough to build up a sand-like surface. The new diameter is tabulated; the measured rough diameter was 2.4897 in. The coefficient is based on the diameter when new. Similar results on nozzle roughness have been reported elsewhere (3).

Tests on two of the nozzles for in-line and free-discharge operation indicate no appreciable difference in coefficient. This corresponds with the data obtained on the long-radius nozzles.

The comparison of the test coefficients with the values predicted on the basis of methods in the "Regeln" (6) shows satisfactory correlation except for nozzle HN-7. The difference is less than 1 per cent which is within the tolerances specified for service rough pipes. The coefficient for nozzle HN-7 is about $2^1/_2$ per cent high. The reason for this difference has not been ascertained.

SUMMARY

The conclusions stated here apply to the results of this limited series of investigations with water as the fluid. Further work and comparison with data obtained by other laboratories may modify these statements.

1 For nozzles of the same area ratio with free-discharge operation with corner taps, there is no consistent difference in coefficients for the size range used in a given type of pipe. The service-rough or black pipe produces coefficients slightly higher than those corresponding to galvanized pipe.

2 Using corner taps, there is no appreciable difference in coefficients for free-discharge and in-line operation.

3 The head measured at an upstream pressure-tap location is lower than the corresponding head at the corner tap. Graphs and data present the relationships. The pipe size and roughness affect the head ratio; the ratio increases as the pipe relative

roughness increases, the amount of the increase being dependent on the diameter ratio.

4 Nozzle coefficients are appreciably reduced for installations having less than 10 diameters of straight pipe between a single elbow and the meter.

5 The nozzle coefficients become larger as the roughness of the pipe increases.

6 All other factors remaining constant, deposits or roughness on the nozzle surface change the coefficients as indicated in Fig. 9.

7 All except one of the I.S.A. nozzles constructed by the laboratory gave coefficients within the tolerance limits specified by the "Regeln" (6).

8 Diameter ratios greater than about 0.82 are not recommended as the meter is supersensitive to pipe roughness, nozzle surface conditions, and the particular installation.

Acknowledgments

The work reported in this paper was made possible through a grant from the A.S.M.E. Special Research Committee on Fluid Meters, assistance in experimental observations by the National Youth Administration workers, and computation assistance from Works Progress Administration personnel. Professor M. P. O'Brien acted in an advisory capacity representing the Fluid Meters Committee.

BIBLIOGRAPHY

1 "Research on Flow Nozzles," by H. S. Bean, *Mechanical Engineering*, vol. 59, 1937, pp. 500–503.

2 "Determining Flow Nozzle Contours at the National Bureau of Standards," by F. C. Morey, *Instruments*, vol. 10, June, 1937, pp. 157–158.

3 "Some Results From Research on Flow Nozzles," by H. S. Bean and S. R. Beitler, Trans. A.S.M.E., vol. 60, April, 1938, paper RP-60-3, pp. 235–244.

4 "Modified I.S.A. Orifice With Free Discharge," by M. P. O'Brien and R. G. Folsom, Trans. A.S.M.E., vol. 59, 1937, paper RP-59-1, pp. 61–64.

5 "Rules for Measuring the Flow of Fluids by Means of Nozzles and Orifice Plates," I.S.A. Bulletins Nos. 9 and 12, 1935.

6 "Regeln für die Durchflussmessung mit Genormten Düsen und Blenden," Deutscher Industrie Normen (DIN), No. 1952, IV Auflage, V.D.I. Verlag, Berlin, Germany, 1937.

7 "Manometer Errors Due to Capillarity," by R. G. Folsom, *Instruments*, vol. 9, 1936, pp. 36–37.

8 "History of Orifice Meters and Calibration, Construction and Operation of Orifices for Metering," Report of the Joint A.G.A.-A.S.M.E. Committee on Orifice Coefficients, The American Society of Mechanical Engineers, New York, N. Y., 1935, pp. 31–38.

Discussion

Howard S. Bean.[4] As stated at the beginning of this paper, the series of tests reported forms a part of the research program on flow nozzles sponsored by the Special Research Committee on Flow Nozzles. It is appropriate to add that the tests made at the University of California were the only ones using water as the fluid, where the nozzle was placed at the end of the pipe, thus giving free discharge into the air. In connection with this program there were a few other tests made at the Ingersoll-Rand plant with nozzles on the end of the pipe to provide free discharge, but in these the fluid was air.

From the standpoint of this particular feature, the second conclusion reported by Dr. Folsom is of particular interest. Except that corner taps were used more often at the University of California than the pipe-line tap locations, there seems to be no reason why this conclusion should be limited to corner taps. The results of the tests with air at the Ingersoll-Rand plant, as well as some earlier tests made by the National Bureau of Standards[5] agree very well with those which were made on the same nozzles within a pipe.

Author's Closure

In discussing the second conclusion of the paper, Mr. Bean calls attention to other results which demonstrate the same correspondence of free and submerged discharge coefficients. This conclusion is generally accepted when a single fluid is present, but some uncertainty exists when a free jet of liquid discharges into a gas. The surface tension has some effect, the magnitude of which is imperfectly understood. The results reported in this paper indicate that this effect is negligible for water discharging into air for the nozzle sizes investigated. The author concurs in the opinion that the pressure tap location has no influence on this conclusion.

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[4] Chief, Gas Measuring Instruments Section, United States Bureau of Standards, Washington, D. C. Mem. A.S.M.E.

[5] "Discharge Coefficients of Square-Edge Orifices for Measuring the Fow of Air," by H. S. Bean, E. Buckingham, and P. S. Murphy; Research paper No. 49, United States Bureau of Standards, vol. 2, 1929, p. 561.

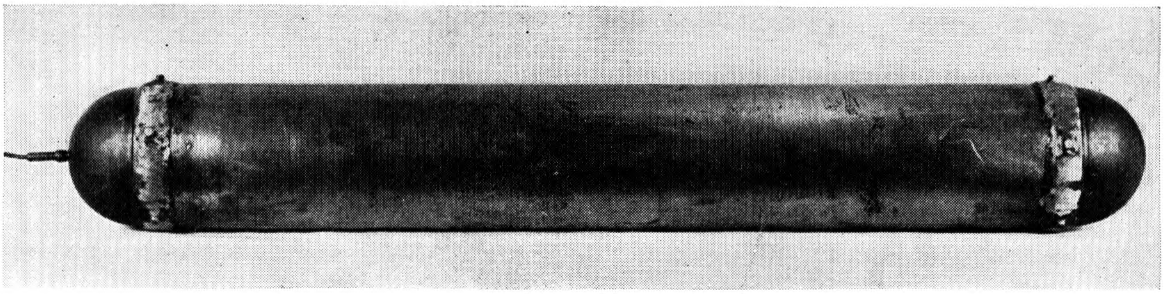


FIG. 1 SPECIMEN TESTED

Creep in Tubular Pressure Vessels

BY F. H. NORTON,[1] CAMBRIDGE, MASS.

The paper is a report of tests on the circumferential and longitudinal creep, under internal pressure and high temperature, in steel tubes such as are used in boilers and oil stills. A description of the testing apparatus is given and the results are shown in the form of tables and curves comparing the tubular creep with the creep of tensile specimens at the same stress.

The tentative conclusion is reached that the circumferential creep is about half that of tensile specimens. The longitudinal creep is negligible.

THIS RESEARCH was started to determine the creep properties of cylindrical pressure vessels as compared with the creep properties of tensile specimens of the same steel, at the same temperature, and under a stress equal to the circumferential stress in the cylinder. Previous work[2] has indicated on a theoretical basis that metal under combined stresses has a lower rate of flow than when under a pure tensile stress of the same magnitude, due to the reduction in shear. Some experimental work[3] has confirmed this and some has shown close agreement between the flow under tensile conditions and the flow under combined stresses. It was, therefore, desired to make careful creep tests on closed pressure vessels of the size and material comparable with tubes used in commercial installations, such as superheaters and oil stills. In the fall of 1936 this research was turned over to the Massachusetts Institute of Technology and work was at once started on the design of the apparatus to test such specimens.

SPECIMENS

The specimen shown in Fig. 1 consists of a carefully machined steel tube with welded-on hemispherical ends and a pressure tube welded to the lower end. The specimen contained a filler which allowed only $^1/_{16}$ in. of free space inside the walls for the gas used to provide the pressure. A cross section of a specimen is shown in Fig. 2 in place in the furnace. The dimensions of the specimen selected by the committee were: over-all length, $28^3/_4$ in., outside diameter, 4 in., and wall thickness, $^3/_8$ in. Whether or not the length is sufficient to eliminate entirely end effects in the center is not certain.

Two types of steel were used in these specimens, one a carbon-molybdenum steel and the other a 4 to 6 per cent chrome-molybdenum steel. The characteristics of these steels have been given in a report prepared by metallurgists of The Babcock and Wilcox Company.[4]

THE FURNACE

A drawing of the furnace is shown in Fig. 2. The base plate (1) supports an 18-8 tube (2) which carries the heating elements. This tube is closed at the ends by the disks (3) of K-30 insulating refractory, and the whole furnace is then covered with 1 in. of high-temperature pipe covering (4). Over this is placed a layer of aluminum foil (5) to reduce the radiation. The furnace tube is wound with five layers of mica tape (6) and then over this is wound No. 14 Kanthal A wire with separate end coils in order to permit a uniform temperature over the specimen. The specimen itself has lugs (8) welded to the outside to guide it centrally in the furnace tube, and the weight is taken by the supporting frame (9) which is carried on separate leveling screws so that the specimen can be accurately adjusted in the furnace.

Stainless-steel bolts (10) connect the crosspiece (11) with the upper end of the furnace tube. This crosspiece is connected in turn to the stationary platform (12) by the support (13) and the horizontal springs (14), thus holding the top of the furnace in alignment with the upper platform. A strut (20) resting in a hardened socket (19) screwed into the platform (11) transmits the expansion of the furnace tube to the lever (15) which is supported by a thin-strip knife-edge (17) between the supporting block (16) and the clamp (18). The lever (15) therefore magnifies the motion of the furnace tube approximately 50 times, thereby opening and closing the contacts (21) as the temperature changes. The lower contact can be adjusted by the knurled head (22) passing through the bushing (23) in the insulating block (24). The beam itself is insulated by fiber strips (14). The upper platform is kept at a constant distance from the base plate by two fused-quartz

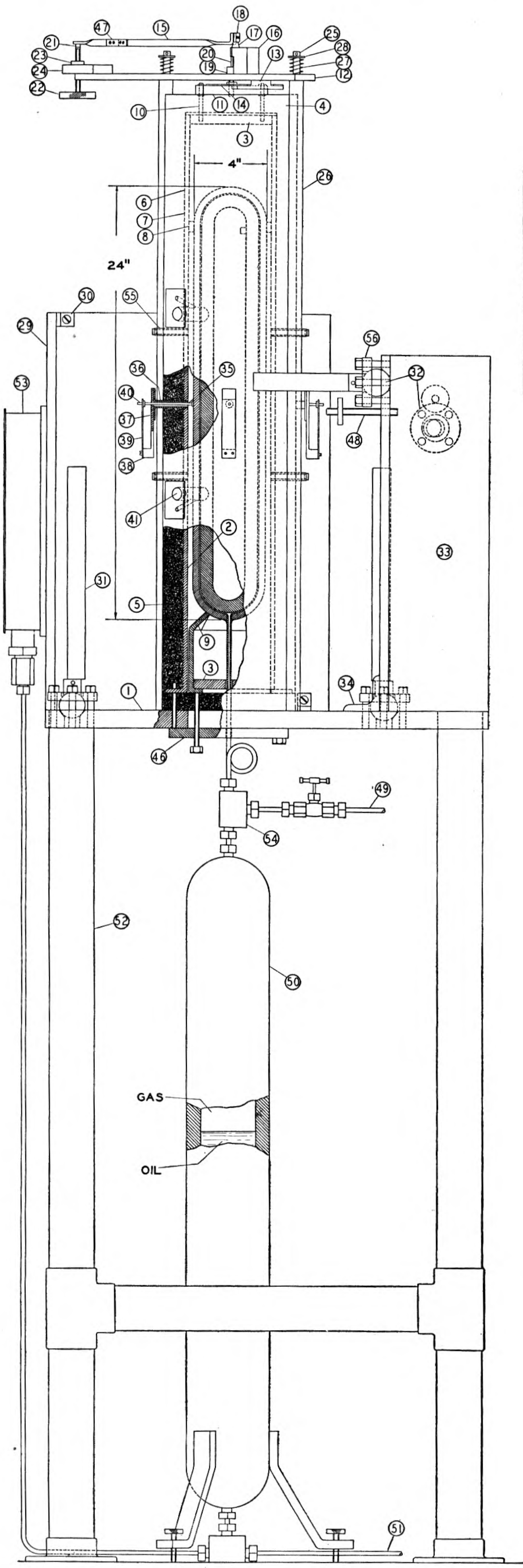
[1] Department of Metallurgy, Massachusetts Institute of Technology.

[2] "The Utilization of Creep-Test Data in Engineering Design," by R. W. Bailey, Proceedings of The Institution of Mechanical Engineers, vol. 131, 1936, pp. 131–149.

[3] "Investigation of Creep and Fracture of Lead and Lead Alloys for Cable Sheathing," by H. F. Moore, B. B. Betty, and C. W. Dollins, Engineering Experiment Station, University of Illinois, Urbana, Ill., Bulletin No. 102, vol. 35, Aug. 19, 1938.

Contributed by the Joint A.S.M.E.-A.S.T.M. Research Committee on the Effect of Temperature on the Properties of Metals, and presented at the Annual Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, held in New York, N. Y., December 5–9, 1938.

[4] "Digest of Properties of Carbon and Alloy Steel Tubing for High-Temperature and High-Pressure Service," Technical Bulletin No. 16, The Babcock and Wilcox Co., New York, N. Y., 1938.



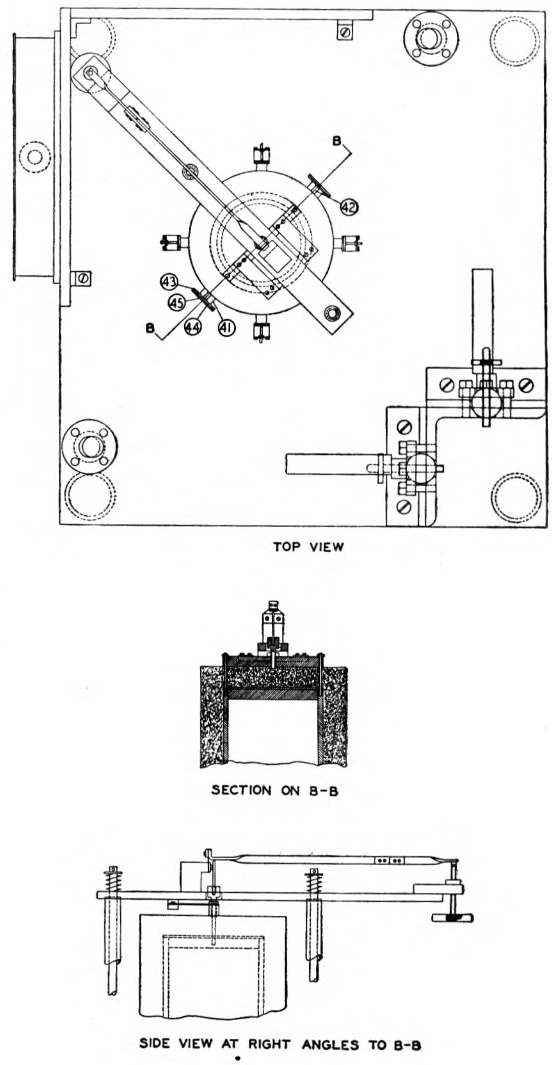


Fig. 2 Furnace and Measuring Equipment

tubes (26) which are held in slight compression by the long bolts (25) and the springs (27).

The principle of operation of this temperature controller has been previously described[5] and need not be discussed in detail. It might be said, however, that this method has proved particularly reliable, and the temperature generally stays within ± 1 F. The temperature of the specimen is measured by four thermocouples, one each on the ends of the gage length and two in the center at opposite diameters.

The measurement of increase in diameter is carried out by employing small fused-quartz tubes (35) which rest on a platinum fixture welded to the surface of the tube in the form of a small tack. The outer end of the fused-quartz tube is supported by a cantilever spring (39) attached to the supporting block (38) which keeps a slight pressure on the fused quartz all the time to hold it firmly on the specimen. A reference mark (40) exactly 5 in. from the center of the specimen is used as a sight for the same pair of telescopes used in the 10-in. tensile test. A small alloy tube (36) welded into the wall of the furnace serves to protect the quartz tubes from injury. The telescopes for measuring the diameter of the specimen are supported on a horizontal post (32) which has a ball-and-socket joint as well as an eccentric mounting to allow accurate alignment of the telescopes at the beginning of the test. A fine screw (48) adjusts the telescopes horizontally. Both telescope posts for horizontal measurements are mounted on the heavy angle (33) in order to give a solid support.

The vertical length measurements are made with the same telescopes supported on the post (31) which also has a ball-and-socket joint and an eccentric mounting for complete alignment. The sighting of the telescopes is carried out through the stainless-steel tubes (41) with the plate (43) welded on the face of them. On this plate is a glass window (44) held by a brass frame (45). An angle tube (42) is used to project a light beam on to the specimen at the point of sighting. The automobile headlight lamps used for this purpose are not shown on the drawing. The reference mark on the specimen is a platinum wire, spot-welded to the surface, which is our usual practice with the tensile specimens. Fine V marks on the wire serve as reference points.

As it would be impractical to employ a small pump in the laboratory capable of producing gas pressures of 10,000 psi for these tests, it was thought advisable to use bottled nitrogen gas up to a pressure of 2000 psi and then compress this gas to the desired amount with an accumulator (50) which has a connection to a small hand-operated hydraulic pump (51) capable of forcing oil into the bottom of the accumulator and thus compressing the gas ahead of it. The connection (49) goes to the tank of nitrogen through the tee (54) which connects with a short capillary tube to the specimen itself. Due to the small volume inside the specimen it is possible to compress the gas from 2000 to 10,000 psi quite readily by this method. Thanks to Dr. Keyes of the chemistry department fittings and valves were constructed which were absolutely tight at pressures up to 10,000 psi. Under no conditions did any leaks occur in the system except through the porous metal of the ends of some of the specimens themselves. This was because the cup-shaped ends were turned out of a billet with considerable longitudinal strain. The pressure itself is read by the gage (53) connected to the bottom of the accumulator. The pressure gages were calibrated before and after the test, and while some hysteresis of the spring was noted, the error was not sufficiently large to be taken into account. A second standard gage was also on hand for periodic checking of the pressure. The whole furnace is supported on a pipe-leg frame (52) to bring it up to a convenient height for reading the telescopes.

Two units of this type were constructed and set up in a separate research room especially devoted to this test, as shown in Fig. 3. As it was necessary to have constant room temperature in order to maintain a constant pressure and constant temperature of the furnace, a room-temperature controller was constructed in connection with a powerful ventilating fan which maintained the room at 85 F with variations of not more than ± 2 F. As this controller is of a simple thermostatic type, a description of it is not necessary here.

[5] "A New Device for Creep Testing," by F. H. Norton and J. A. Fellows, *Metal Progress*, vol. 24, 1933, p. 41.

Method of Making the Test

The tubular test specimen was first placed in a jig and the positions of the reference marks and strut attachments were carefully laid out with a fine scratch on the surface of the specimen. It

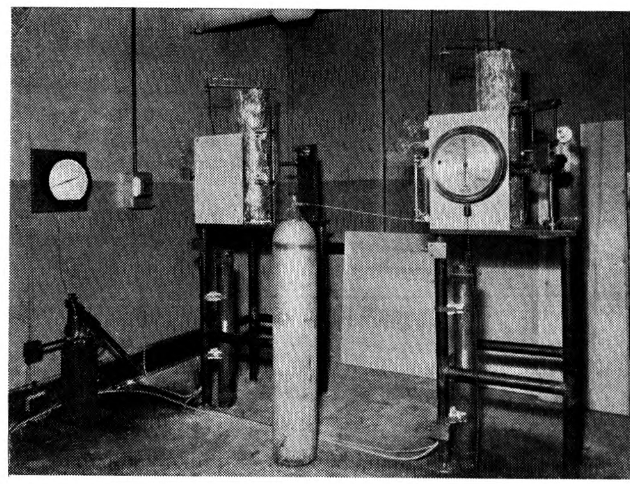


Fig. 3 Testing Laboratory With Two Furnaces

was then taken over to a welder and the platinum attachments spot-welded in place in their correct location. The thermocouples were next put in place and bound to the sides of the specimen with Chromel wire. The thermocouple wires themselves were insulated with porcelain beads to the end of the furnace and then with asbestos tubing. The connecting nipple was then soldered on to the capillary tube with soft solder, which completed the work on the specimens.

The placing of the specimen in the furnace was a rather delicate operation and it was found advisable to raise the furnace itself with a tackle supported from the ceiling and place the specimen on its supporting ring with another tackle, allowing the capillary tube and the thermocouple to pass down through the bottom of the furnace. The specimen was then adjusted in a vertical position and held there temporarily while the furnace itself was lowered carefully around the specimen, and clamped down with the three bottom bolts. Great care had to be taken during this operation to prevent dislodging the platinum reference marks, and it was found advisable to wrap the center of the specimen with soft twine which protected the platinum marks and which later burned off.

After the furnace itself had been carefully leveled, the specimen was adjusted for height and rotary position as well as possible, but the silica struts were not put in place at this time.

The furnace was heated up gradually, taking a period of several days to reach the equilibrium value. Then, during a period of several weeks, the current through the end coils was adjusted to get a satisfactory temperature distribution. Because of the heavy specimen it was necessary to wait at least 24 hr after each adjustment before the next one could be made. After the temperature had become satisfactorily balanced, the height of the

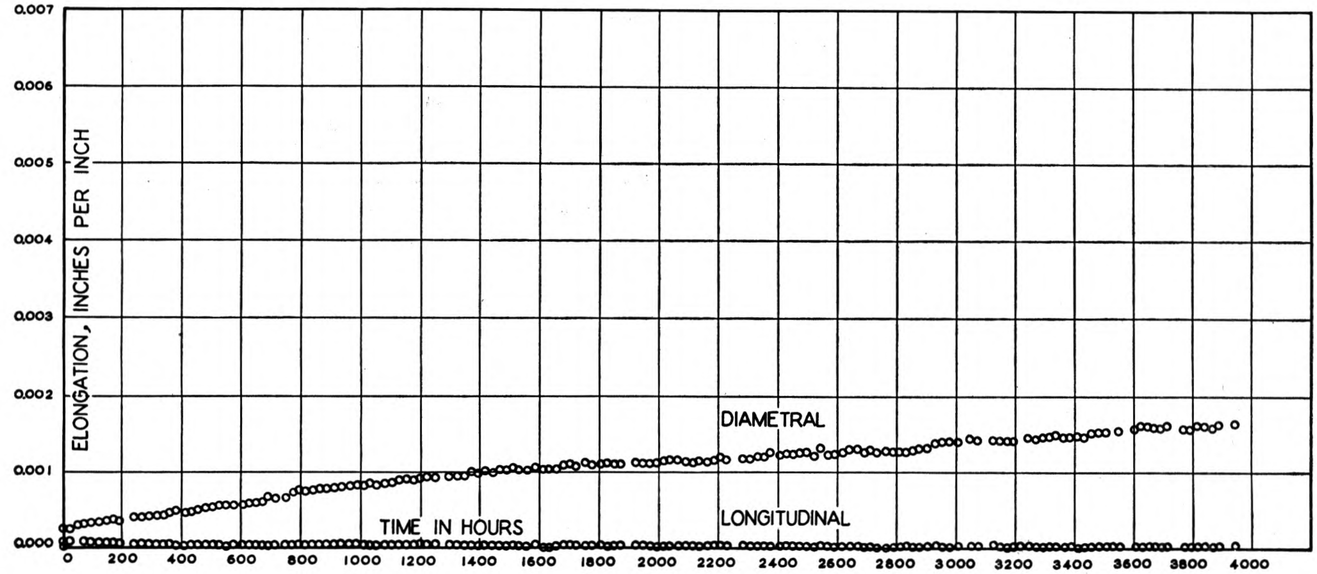


Fig. 4 Tubular Creep Test
(C-Mo steel at 1050 F and 1617 psi.)

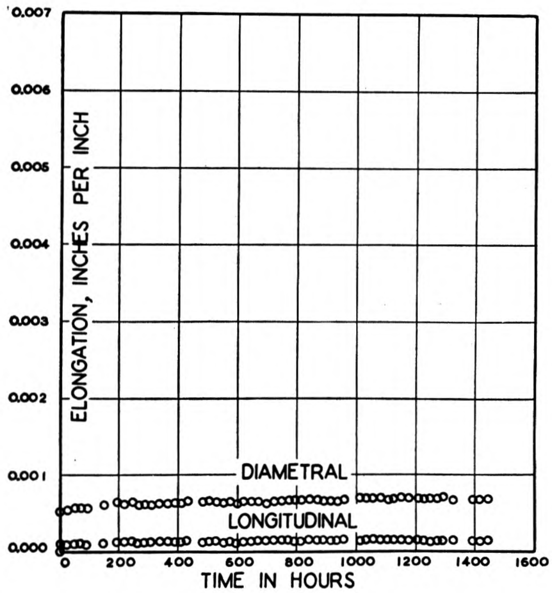


Fig. 5 Tubular Creep Test
(C-Mo steel at 800 F and 4000 psi.)

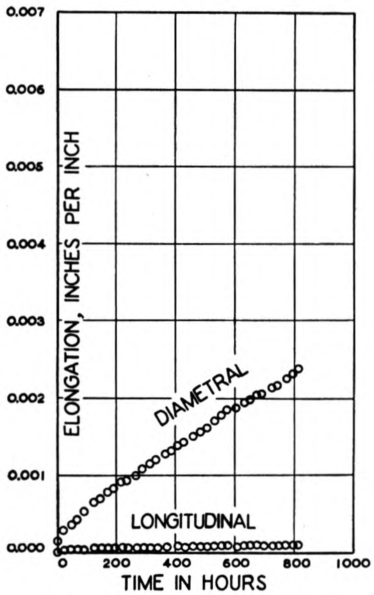


Fig. 6 Tubular Creep Test
(4–6 Cr steel at 1200 F and 650 psi.)

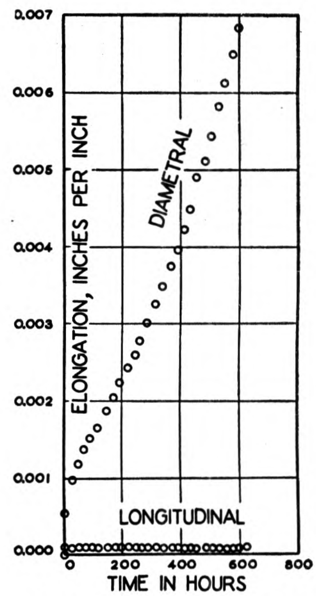


Fig. 7 Tubular Creep Test
(C-Mo steel at 1050 F and 3230 psi.)

specimen was finally adjusted with the leveling screws and the silica struts were put in place. The telescope posts were then lined up to give simultaneous forms on the two marks.

Length measurements were taken with the telescopes with no pressure, then the pressure was brought up to the working value as quickly as possible. At the lower pressures this could be done in a few minutes, but at the higher pressures it took from 6 to 8 hours' pumping to reach the desired value. Another length measurement was taken with the telescopes immediately after the correct pressure was established, and readings were taken thereafter every 24 hr. All the readings were taken by two observers and the results averaged.

In case a specimen showed a drop in pressure due to leakage, the pressure was pumped up to the correct value every morning and every night. In some cases the leakage was small; in one case there was none at all, and in one case leakage was considerable. Except in the two cases where leakage was severe the pressure was held within ± 1.0 per cent at all times.

TABLE 1 FINAL PRESSURES AND STRESSES

Steel	Temperature, F	Internal pressure, psi	Circumferential stress, psi			Longitudinal stress, psi
			f_i[a]	f_o[b]	f_{av}[c]	
C-Mo	800	4000	19430	15430	17230	7720
C-Mo	800	9240	44890	35650	39792	17830
C-Mo	1050	1620	7870	6250	6970	3120
C-Mo	1050	3230	15690	12460	13910	6230
4–6 Cr-Mo	1200	650	3160	2510	2800	1250
4–6 Cr-Mo	1200	920	4490	3568	3980	1780

[a] f_i = circumferential stress on inside of cylinder.
[b] f_o = circumferential stress on outside of cylinder.
[c] f_{av} = circumferential stress, average.

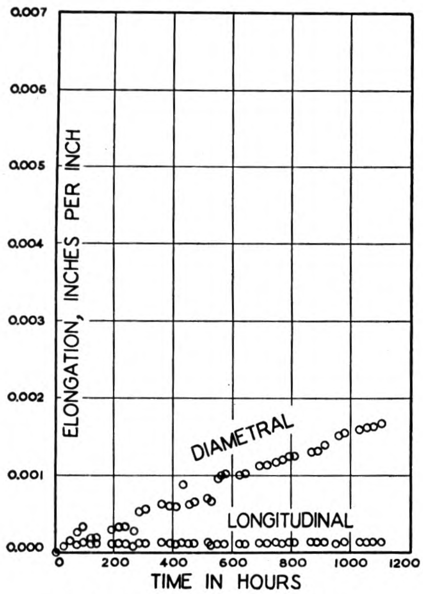


FIG. 8 TUBULAR CREEP TEST
(4–6 Cr steel at 1200 F and 923 psi.)

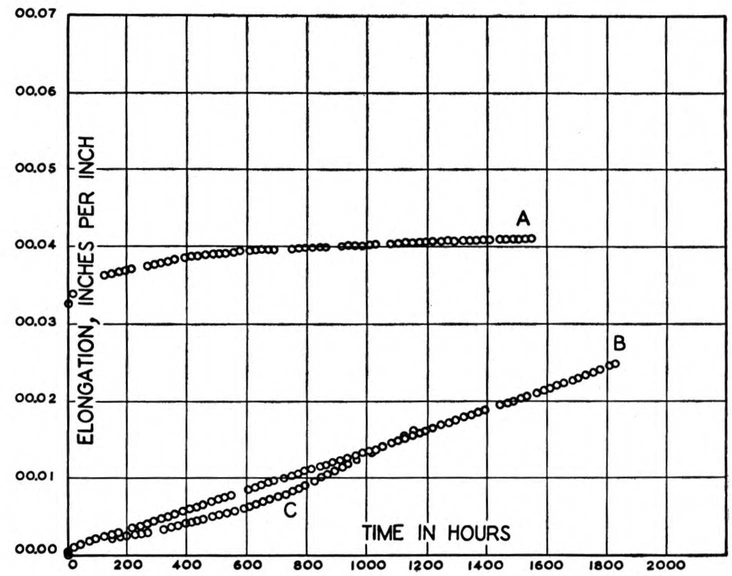


FIG. 9 TENSILE CREEP TEST

A = C-Mo steel....................	800 F	39,170 psi
B = 4–6 Cr steel....................	1200 F	3,982 psi
C = C-Mo steel....................	1050 F	13,933 psi

CONDITIONS OF TESTING

The temperatures selected by the committee for these tests were 800 and 1050 F for the carbon-molybdenum steel, and 1200 F for the 4–6 chrome-molybdenum steel. The duration was to be 1000 hr. The pressures were selected, as nearly as could be determined from past experience on these materials, to give circumferential creep rates of 5 and 50 per cent per 100,000 hr. The loads were altered somewhat after the first test under a given condition in order to adjust the second load more nearly to the conditions wanted. The final pressures and stresses in the metal figured out by the Lamé formula by The Babcock and Wilcox Company are given in Table 1.

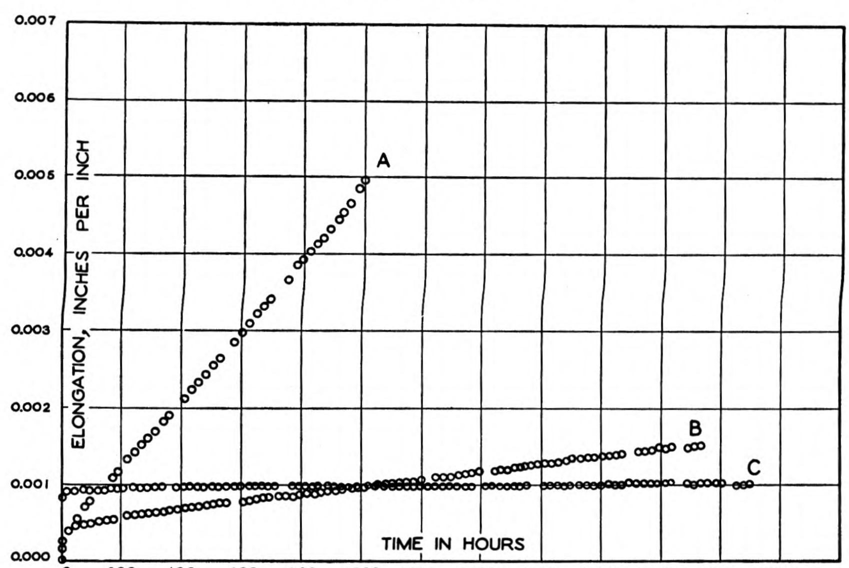


FIG. 10 TENSILE CREEP TESTS

A = 4–6 Cr steel....................	1200 F	2,818 psi
B = C-Mo steel....................	1050 F	6,973 psi
C = C-Mo steel....................	800 F	17,226 psi

PRECISION OF THE MEASUREMENTS

The readings of the longitudinal creep are accurate to two parts in a million, which is equivalent to the regular 10-in. tensile creep tests. The creep in diameter, however, is good only to six parts in a million, because the gage length is only $^1/_3$ as great. In regard to the temperature, the distribution was good to ± 3 F over the gage length, and the variation in temperature from day to day was ±1 F.

Micrometer readings of the diameter in the center of the specimen were made before and after each run. In every case the permanent increase in diameter determined in this way agreed with the telescope readings.

The first specimen was run at 1200 F and initially some difficulty was had in adjusting the silica struts to give an even bearing so that consistent readings could be obtained, but toward the middle of the run this difficulty was eliminated and satisfactory readings were obtained. However, due to the rather rapid leakage of gas through the ends of the specimen it was necessary to pump up the pressure frequently, which gave a condition of alternating stress causing the results to be out of line with the later tests. Therefore, the results are shown merely to indicate the improvement in technique during the first run. The specimen run at 800 F and 9240 psi also leaked so badly that creep rate is only an approximation. The flow curves in both a circumferential and longitudinal direction are shown in Figs. 4 to 8.

It will be noted first that the longitudinal creep is rather low in most cases and sometimes has even a negative value. The circumferential creep gives a curve quite similar to the usual tensile creep curves except there seems to be a tendency to take a longer period of time in coming to an equilibrium rate, which is due, perhaps, to adjustment of stresses in the tubular specimen. Therefore, there is some possibility that the rates given in the first runs may be a little higher than the equilibrium rate even though the tests were run as long as is generally considered necessary in the tensile test. The last specimen run was carried along for a considerably greater period as requested by the committee at H. J. Kerr's suggestion and we find here a definite decrease in the rate after a period of around 2000 hr, a change which might have occurred in the other specimens had they been carried through this period.

Tensile Tests

Specimens were cut out of the walls of the same batch of tubing that the pressure vessels were made from and these were run in the regular 10-in. furnace as a standard tensile creep test. These were cut with their length parallel to the axis of the tube. The creep curves for these runs are shown in Figs. 9 and 10 and need no special comment.

Comparison of the Creep Rates in the Tubular and Tensile Specimens

Log-log plots of the circumferential creep rate, based on the average stress, and the creep rate in tension are shown in Fig. 11,

TABLE 2 TUBULAR SPECIMENS

Steel	Temp, F	Internal pressure, psi	Circumferential creep per 100,000 hr, per cent	Longitudinal creep per 100,000 hr, per cent	Time interval from which rate is computed, hr
C-Mo	800	4000	0.7	0.6	400–1500
C-Mo	800	9240	10.0	0.0	200–400
C-Mo	1050	1620	5.8 } 2.2 }	0.0	200–1200 1800–3900
C-Mo	1050	3230	139.0	0.0	400–800
4–6 Cr-Mo	1200	650	25.0	0.9	200–800
4–6 Cr-Mo	1200	920	15 ?	0.0	600–1100

TABLE 3 TENSILE SPECIMENS

Steel	Temp, F	Load, psi	Rate of creep per 100,000 hr, per cent	Time interva from which rate is computed, hr
C-Mo	800	17230	0.3	400–2300
C-Mo	800	39170	22.0	800–1500
C-Mo	1050	6970	4.8	600–2100
C-Mo	1050	13930	195.0	800–1200
4–6 Cr-Mo	1200	2818	46.0	400–1000
4–6 Cr-Mo	1200	3980	137.0	1000–1800

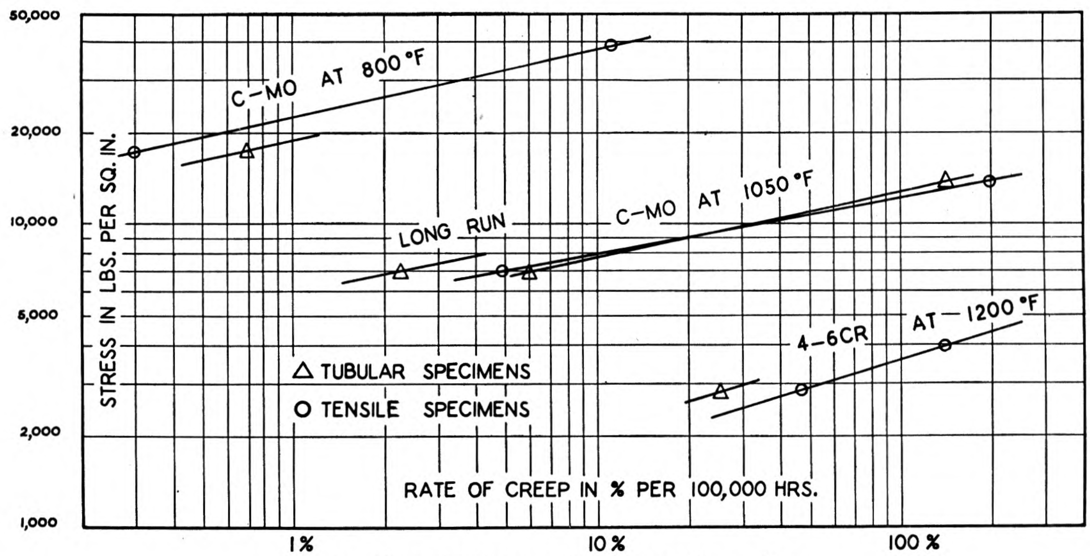


Fig. 11 Circumferential and Tensile Creep Rates

while the specific data on the creep rates are tabulated in Tables 2 and 3.

The comparison between the circumferential creep of the tubular specimens, and the tensile creep would indicate that at 800 F the tubular specimens creep faster under the same stress. At 1050 F there is little difference between the two conditions, while at 1200 F the tube creeps somewhat more slowly than the corresponding tensile specimen. The differences, however, are not greatly in excess of the experimental error and it would look in general as if the rates were substantially the same under the two conditions. However, if the tubular specimens are run for a longer period of time, which was done in the case of the carbon-molybdenum steel at 1050 F, a material decrease in the rate occurs from 5.8 per cent at the end of 1000 hr to 2.2 per cent from 2000 to 4000 hr. It would, therefore, seem, at least from this single test, that the tubular specimens take a much longer time to reach an equilibrium rate of flow than the tensile specimens. It is probable that if all of the tubular specimens had been run for from 4000 to 5000 hr the rates would have been materially lower. Some work carried out by Prof. H. F. Moore[3] on the flow of lead tubes confirmed this conclusion, as he states that the tubular specimens required a considerably greater length of time to reach an equilibrium flow than the tensile specimens. If this point of view is correct then it may be stated that the tubular specimens creep at about half the rate of the tensile specimens under the same stress.

It should be realized that it is improper to base any general conclusions on the few tubular tests carried out in this project. More data are needed to supplement the present information in order to formulate definite factors for converting creep rate in tension to creep rates of tubes under internal pressure. Especially, tests should be made with other diameters and wall thicknesses.

The longitudinal creep is in most cases small; sometimes it is slightly positive and sometimes slightly negative. It is probably safe to assume in any design problem a zero change in length of the tubular member during its life.

Acknowledgments

The author wishes to express his appreciation for the assistance received in this investigation by members of the Joint High Temperature Committee. The assistance of The Babcock and Wilcox Company, and especially H. J. Kerr, in the difficult problem of making up the test specimens and for advice and suggestions during the test are especially appreciated. Also, the help and suggestions of C. E. MacQuigg, chairman of the committee, have been most helpful. The work of John Fellows in setting up the apparatus and Donald Fellows in making the observations has been of the greatest assistance in carrying out these tests.

Discussion

C. O. Rhys[6] and M. S. Northup.[7] The purpose of the investigation is of extreme interest because of the high-temperature high-pressure equipment now in service and the certainty that temperatures and pressures of future equipment will increase rather than decrease.

The tests reported by the author carry on more completely and fully a line of investigation touched briefly by tests at room temperature made in our laboratory in 1933 on $3^1/_2$-in. OD $\times$ $^1/_4$-in. wall lead tubes to determine qualitatively any difference between creep in tension specimens and creep in tubes. It is of interest that these tests showed that in closed-end tubes the axial creep is negligible, a conclusion drawn by the author from his test results. In this connection R. W. Bailey,[8] Metropolitan-Vickers, Ltd., England, also found that axial creep need not be considered.

The range of creep rate generally used in design work is from 0.1 to 10 per cent per 100,000 hr. An examination of the tables for tubular and tensile specimens shows that most of the deformation rates on the specimens were outside this range and the writers are reluctant to discuss the results. These enormous creep rates may and probably do have a considerable effect on the creep-rate distribution through the wall.

According to experiments made by R. W. Bailey with a lead cylinder $3^1/_4$ in. OD and $^1/_8$ in. thick the diametral creep rate with no axial stress is from three to four times as much as with the axial stress set up by the internal pressure.

From experiments made in our own laboratories with a lead cylinder $3^1/_2$ in. OD and $^1/_4$ in. thick, the ratio is between 1.5 and 2.5 to 1. The theory for a very thin cylinder would give about 5 to 1. It is interesting to note that as the cylinders become relatively thinner the theoretical result is approached as shown in Table 4.

According to Bailey's experiments for these cylinders, the diametral creep rate is a minimum, when the axial stress is that due to internal pressure. It would appear from this that most pressure vessels are subjected to the least possible amount of diametral creep. By treating the $3^1/_2 \times 3$-in. lead tube tested in our laboratory as a thick cylinder, the ratio of diametral creep rates at the outside surface, with and without axial pressure stress, is 1 to 2.57. This is a good check with the 2.5 found experimentally. For the steel cylinder of the paper the calculated ratio is 1.42 which is a close check with $195/139 = 1.4$.

TABLE 4 RATIOS OF DIAMETRAL CREEP WITH AND WITHOUT THE AXIAL STRESS CAUSED BY INTERNAL PRESSURE

	OD/ID	Creep ratio
Standard Oil Development Co.	1.17	1.5 to 2.5
R. W. Bailey	1.08	3 to 4
Theoretical	1.00	5

Comparisons of creep rate at the inside wall of tubular specimens would be of great interest as it is this creep rate which governs design of cylindrical pressure vessels.

We agree with the author that the few tests covered by the report are insufficient to arrive at any definite conclusions. It is believed that further work along the same lines is desirable, but with deformation rates of from 0.1 to 10 per cent per 100,000 hr and different ratios of diameter to thickness, so that the results will apply to both pressure-vessel and tube design. Ultimately, for tube design, such a program should include the determination of the effects of heat transfer through the wall if any practical way can be found to do so.

Author's Closure

The comments of Messrs. Rhys and Northup give some interesting data on lead cylinders which seem to check well with the theoretical work of Bailey. We quite agree that some of the creep rates in our work were much higher than would be encountered in service, but due to the fact that reliable tensile values of the particular steels used were not available before the tests on the tubes were started, some of the pressures were set too high.

We agree also that it would be desirable to carry the test with a thermal gradient through the tube wall, but the experimental difficulties of carrying this out are so great that we can see no way of accomplishing it without extremely costly equipment.

[6] Process Design Department, Standard Oil Development Company, Elizabeth, N. J.

[7] General Engineering Department, Standard Oil Development Company, Elizabeth, N. J.

[8] "Utilization of Creep-Test Data in Engineering Design," by R. W. Bailey, Proceedings of The Institution of Mechanical Engineers, vol. 131, 1935, pp. 131–349.

Effect of High Temperatures and Pressures on Cast-Steel Venturi Tubes

By W. S. Pardoe,[1] Philadelphia, Pa.

The author presents results of tests showing the expansion in the main and throat diameters of cast-steel venturi tubes under dry heat and pressure, respectively. From these data values are obtained for the coefficient of expansion and the modulus of elasticity, and the comparative effects of high temperatures and pressures are shown.

The results of temperature tests conducted on a 5.625 × 4.25-in. extra-heavy cast-steel Simplex venturi tube at H. Brinton Company's oven on April 24, 1929, are shown in Fig. 1.

The coefficient of expansion of the throat per deg F is 0.0000108 and that of the main is 0.00000946; a mean value of 0.000010 might be used. This is greater than the usually quoted values for cast steel and about equal to that of bronze.

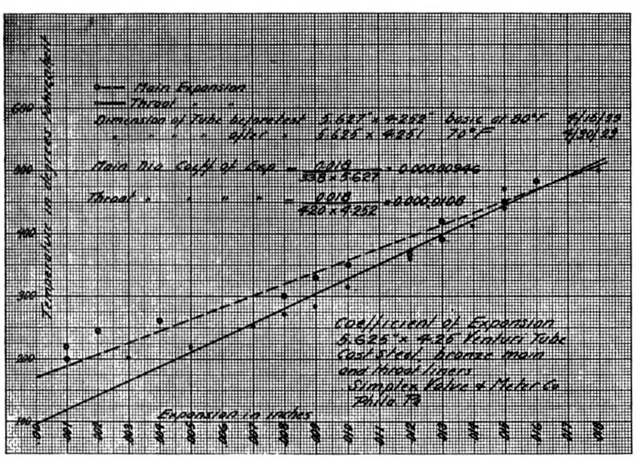


Fig. 1 Expansion of Venturi Tube in Dry Heat

Since the discharge varies directly as the square of the throat diameter, the error resulting from neglecting the expansion is 0.2 per cent per 100 deg F, which is not a negligible quantity when the rise in temperature may be 800 deg or more.

The tests were conducted as follows:

The basic dimensions were taken at a temperature of 80 F. These were main diam 5.627 in., and throat diam 4.252 in. An outline sketch is shown in Fig. 2. The tube was then placed in the oven, the temperature of the oven was raised to 200 F, and held constant for 15 min. The tube was withdrawn, measured in less than 30 sec, and replaced, after which the oven temperature was again raised and held stationary for 15 min. This process was repeated for each temperature.

The high-pressure tests were conducted in the hydraulic laboratory of the civil-engineering department of the University of Pennsylvania on an 8 × 5-in. extra-heavy cast-steel Simplex venturi tube, dimensions of which are shown in Fig. 2.

The results are shown in Fig. 3 and give a modulus of elasticity (defined as unit pressure, in lb per sq in., divided by unit expansion of the throat diameter in in. per in.) of 17,000,000 lb per sq in.

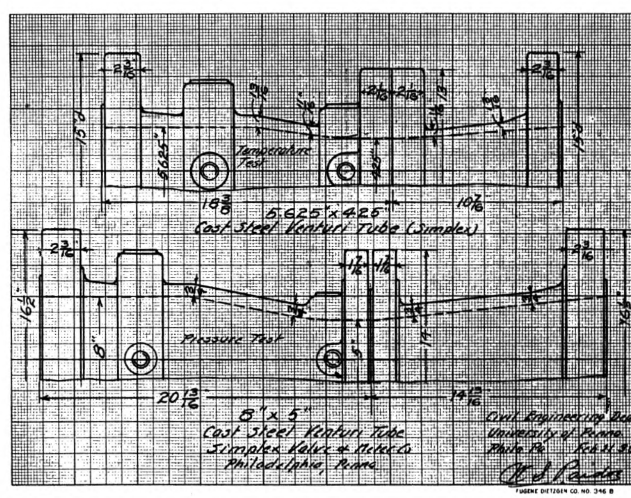


Fig. 2 Dimensions of Venturi Tubes Used in Temperature and Pressure Tests

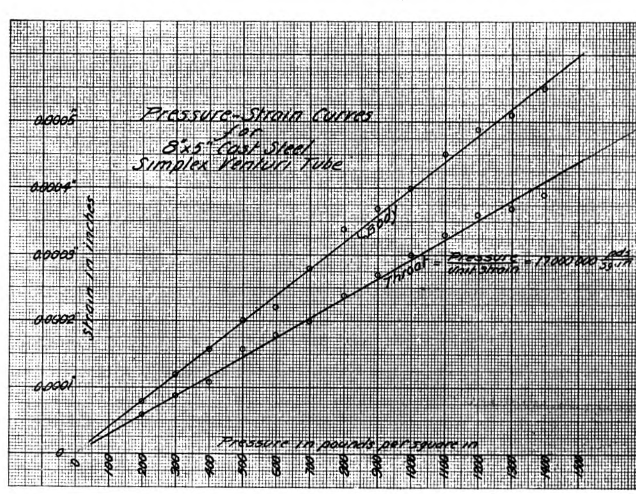


Fig. 3 Pressure-Strain Curves for 8 × 5-In. Cast-Steel Venturi Tube

[1] Professor of Hydraulic Engineering, University of Pennsylvania. Professor Pardoe was graduated in mechanical engineering from the Ontario School of Practical Science and from the University of Toronto in applied science. He spent three years at marine-engine and pump drafting and was employed as hydraulic engineer for three years by the Canada Foundry Company. He has served for twenty-seven years as instructor, assistant professor, and professor of hydraulic engineering at the University of Pennsylvania, and as consulting engineer on hydraulic power, dams, and hydraulic problems.

Contributed by the Special Research Committee on Fluid Meters for presentation at the Annual Meeting of The American Society of Mechanical Engineers, held at New York, N. Y., Dec. 5–9, 1938.

Discussion of this paper was closed January 10, 1939, and is published herewith directly following the paper.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

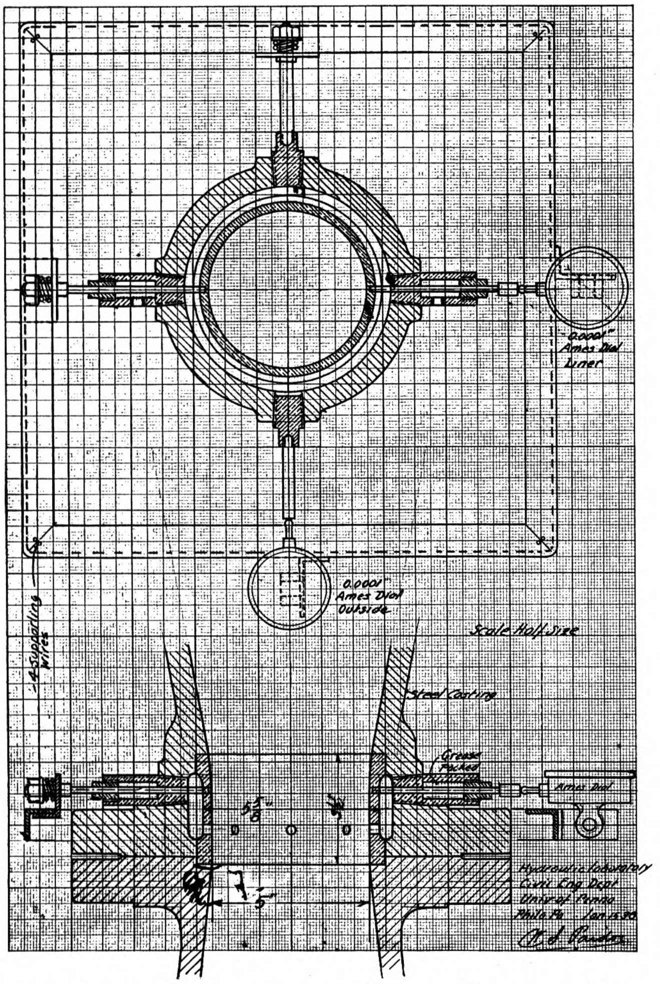


FIG. 4 DETAILS OF APPARATUS FOR MEASURING THROAT AND BODY EXPANSIONS UNDER PRESSURE

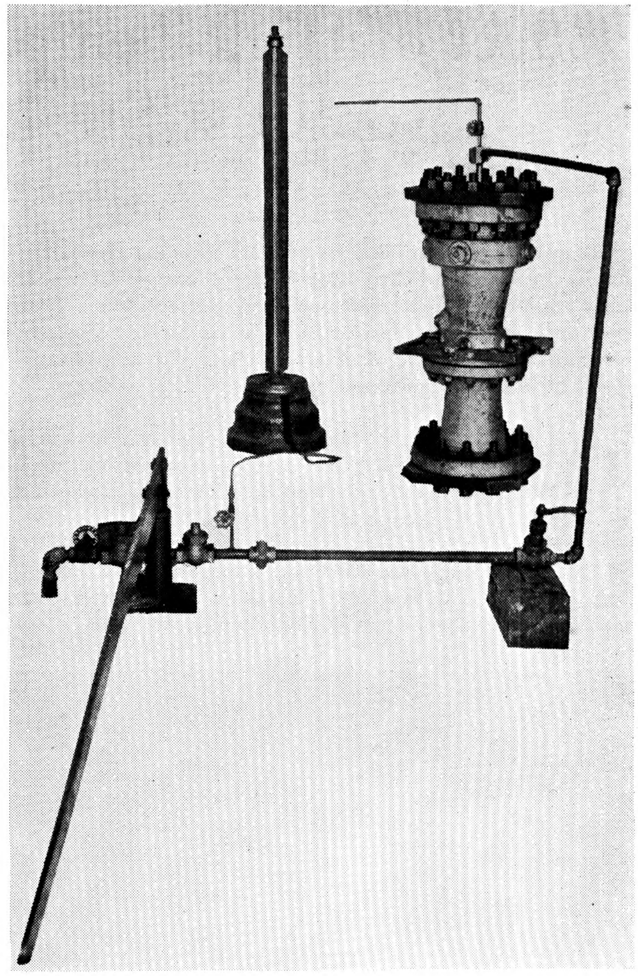


FIG. 5 APPARATUS USED FOR PRESSURE TESTS

Thus there is an expansion of 0.00025 in. for a pressure of 850 lb per sq in., which if neglected would make an error of 0.01 per cent, an entirely negligible quantity.

The apparatus for measuring the throat and body expansions is shown in Fig. 4. Steel rods of $^1/_8$ in. diam pass through grease stuffing boxes and are tapped into the bronze throat liner. A square frame of 1-in. angle iron, hung by wires from the meter flange, support two 0.0001-in. Ames dials which can be read to 0.00001 in. The meter was blank-flanged and water pumped into it with a 1-in. hand pump having a long lever. The pressure was recorded by a 5000-lb pneumercator gage (differential plunger mercury-loaded). The entire apparatus is shown in Fig. 5.

Discussion

S. R. BEITLER.[2] These tests, especially those having to do with the effect of temperature on venturi tubes, are of major importance for accurate measurements of high-temperature fluids, since such tubes are used generally for the measurement of boiler feedwater. It is not surprising that for this particular design of tube the coefficient of expansion approximates that of bronze; since the piezometer ring behind the throat would mean that the bronze throat section was not supported by the steel shell at this point.

Since the description does not tell whether there was a bronze insert at the inlet-pressure connection, it is difficult to predict whether this reasoning would follow for this section.

A paper[3] which was presented to the Society in 1931, reported on a set of similar tests for sharp-edged orifice plates and steel flow nozzles clamped in steel flanges. The tests showed that for these primary elements it was possible to calculate the increase in diameter for the coefficient of expansion of the metal (monel and 0.12 carbon metal) as accurately as it could be measured. It was also found that no change in shape or flatness was caused either in the nozzles or in the orifice plate.

L. K. SPINK.[4] The author does not draw any conclusions regarding the effect on the flow coefficient due to the measured change in dimensions caused by temperature and pressure. It is possible that warping caused by use of dissimilar metals, having different coefficients of expansion, may create effects exceeding those calculated from the changes in dimensions. It has been found that the use of 18 and 8 steel orifice plates in steel flanges at high temperatures in some cases produces undesirable warping. The present engineering trend is toward choosing orifice-plate materials to minimize warping or toward

[2] Assistant Professor of Mechanical Engineering, The Ohio State University, Columbus, O. Mem. A.S.M.E.

[3] "A Study of Primary Metering Elements in 3-In. Pipe," by S. R. Beitler, P. Bucher, and T. C. Barnes, Trans. A.S.M.E., vol. 54, 1932, paper RP-54-3.

[4] The Foxboro Company, Foxboro, Mass.

mounting the plates with means to permit expansion and contraction without warping.

Nevertheless, it is sound practice to correct for all known variables and to eliminate in so far as possible all unknown variables. The writer believes the author would have been justified in concluding that the venturi-coefficient calculations should be based upon the dimensions at operating conditions.

Author's Closure

Professor Beitler's suggestion that the high coefficient of expansion can be explained by a loose throat piece does not seem tenable. The pressure-strain curves indicate that the bronze throat expands, although subject to no difference of pressure, which would indicate that the bronze throat piece was prestressed by force fitting and expanded with the body. Also, there was no bronze liner in the inlet section and it had approximately the same coefficient of expansion, i.e., 0.00000946 main and 0.0000108 throat.

The author welcomes the reference to "A Study of Primary Metering Elements," by Professors Beitler, Bucher, and Barnes. He should have referred to this before rather than after his experimental work, but such is life: We go through it smashing in open doors. The results indicate an appreciable error in using the normal coefficient of expansion (0.0000065) in this particular tube.

Mr. Spink draws attention to the warping of large orifice plates. Professors Bucher and Barnes say there is none. This paper concerns itself with venturi tubes and the whole idea is to determine the corrections to the diameters as measured at normal temperatures and pressures when using the tubes at high temperatures and pressures.

The author desires to take this opportunity to thank those contributing written and oral discussions to his paper.

Twenty Years of Machine Activity in the Woolen-and-Worsted Industry

By A. W. BENOIT,[1] BOSTON, MASS.

The paper is a discussion of data, compiled from various sources and presented in the form of graphs and tables, tracing conditions in the woolen-and-worsted industry with a view to determining what may be expected in the future. Past performance is analyzed with respect to basic equipment, its active working time, the amount of raw materials used, and working hours.

The conclusions are reached that the liquidation of mills as a whole is nearly finished, that present spindle equipments are not excessive, but that there is still considerable excess loom capacity.

During the last twenty years the textile industry has been passing through a critical period, the end of which is not yet in sight. While the depression has been a contributing factor, it was not the principal cause, because the present difficulty made its first appearance about 1923. There has been a buyers' market with rare exceptions for many years, and the textile manufacturer has been at the mercy of the purchaser of his products. He has been unable to make consistent profits, burdensome legislation has dogged his steps, labor unrest has added to his difficulties, raw-stock markets have been unsteady, styles have changed frequently and radically, and, like the caged squirrel, he has gone round and round, getting nowhere.

Within the industry itself there is much hesitation and uncertainty among its members, due somewhat to the rapid political, social, and economic changes which affect its operation. The competition between mills for the business which is available is so sharp that it is destructive, and many liquidations have resulted. This is equally true of both the cotton and wool branches of the industry.

In the eleven years from 1925 to 1936, the number of cotton spindles in the United States dropped from about 38 million to 27 million, or at the rate of a million a year. At the same time, the consumption of cotton by the mills, while it had its usual annual fluctuations, averaged about the same as prior to 1925. An analysis of this situation, made in October, 1937, showed quite clearly that the increased number of hours at which spindles and looms were being operated was the greatest contributing factor and that, if the cotton mills continued to operate 80 hours or more per week, there still would be considerably more machinery in place than would be necessary to supply the country's needs.

A similar analysis of the machinery in the woolen-and-worsted industry was deemed desirable to determine, if possible, the facts as to trends and future possibilities. There are certain data prepared by various federal agencies, as well as those kept by private groups, which are available and from which tables and graphs can be made which show what has occurred. The analysis has been confined, in so far as possible, to that part of the industry which is engaged in making apparel, omitting the carpet and felt manufacturers. These latter are such a relatively small part of the whole that they affect the general picture but little.

Where it has been possible, the data on woolen and worsted machinery have been kept separate, and the reports are so made that this can be done up through the spindles. The looms, however, in many cases weave both woolen and worsted fabrics and it is impossible to make any intelligent division of this machinery.

In order to get a general picture of the industry as it is at present, with reference to its location and size, Table 1 was prepared. Worsted spindles include flyer, cap, and ring spindles, and woolen spindles include both frames and mules.

TABLE 1 UNITED STATES DISTRIBUTION OF WOOLEN AND WORSTED MACHINERY IN 1938

(Information furnished by textile-mill directories and other sources)

Section	Woolen and worsted spindles	Per cent of total	Woolen and worsted looms	Per cent of total
1	3887000	93	52500	92
2	201000	5	3600	6
3	69000	2	900	2
	4157000	100	57000	100

Section 1, north of Maryland and Kentucky, and east of Mississippi River.
Section 2, south of section 1, and east of the Mississippi River.
Section 3, west of the Mississippi River.

It will be noted that the total figures of Table 1 do not check exactly with the data from the U. S. Bureau of the Census, used later in this paper; however, they are sufficiently accurate to show distribution in percentages and, at any rate, they are the only data on this phase of the subject that the author has been able to find.

It is to be noted that the great bulk, over 90 per cent, of this industry is in the northeastern section of the country, including the New England States, New York, Pennsylvania, New Jersey, Delaware, Ohio, Indiana, Michigan, and Wisconsin. Of the total, about 60 per cent is in the New England States and 20 per cent in New York, Pennsylvania, New Jersey, and Delaware. This area is compact and homogeneous, and much alike as to climate, working conditions, labor rates, usual hours of labor, and general competitive conditions such as freight, power, and proximity to raw stock and market.

At the present time a small part of the machinery is in the South, and if this region operates under any different conditions as to hours and rates of labor from the northeastern section of the country, it will have comparatively little effect upon the general situation.

The United States Department of Commerce, Bureau of the Census, reports the monthly activity of woolen and worsted machinery, and its report as of July, 1938, shows the following machinery in possession of mills:

[1] Member of the firm and head of the textile department of Chas. T. Main, Inc. Mem. A.S.M.E. Mr. Benoit was graduated with the degree of B.S. in civil engineering from Tufts College in 1907. He has worked with Chas. T. Main and Chas. T. Main, Inc., since 1909 and in his present position since 1919.

Contributed by the Textile Division and presented at the Annual Meeting of The American Society of Mechanical Engineers, held in New York, N. Y., December 5–9, 1938.

Discussion of this paper was closed January 10, 1939.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

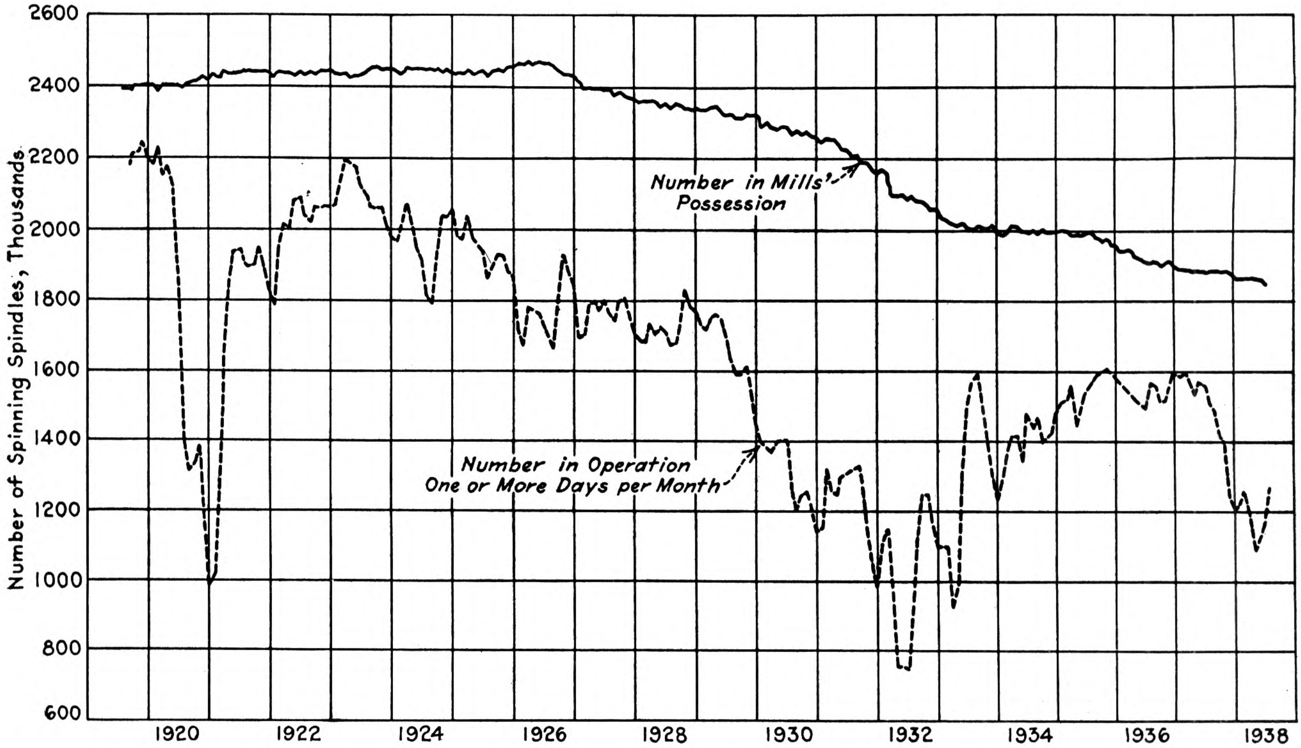


FIG. 1 NUMBER OF WOOLEN-SPINNING SPINDLES

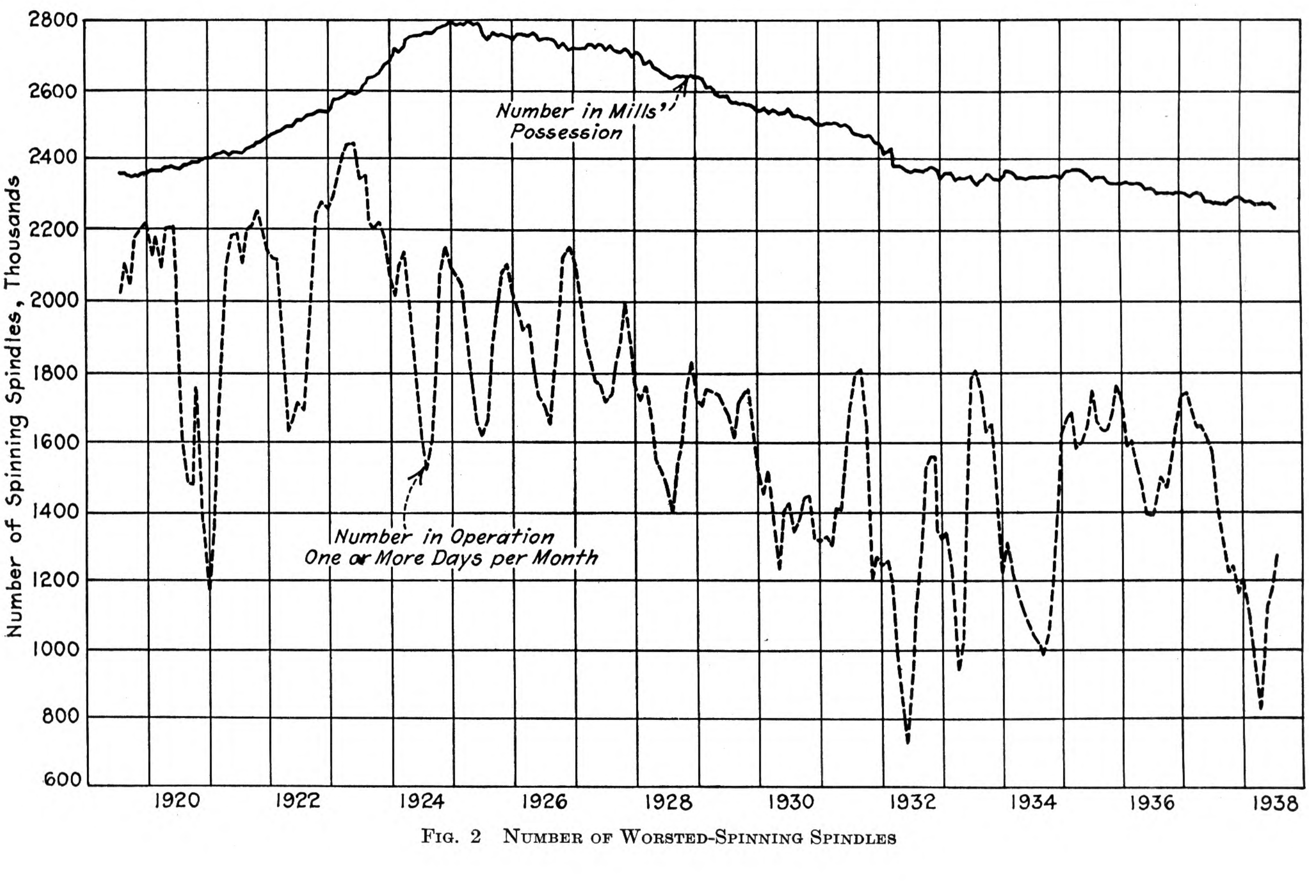


FIG. 2 NUMBER OF WORSTED-SPINNING SPINDLES

Woolen and worsted broad looms[2]	44,887
Woolen and worsted narrow looms	7,948
Woolen-spinning spindles	1,856,844
Worsted-spinning spindles	2,258,045

Of the 44,887 looms listed, 26,307 or 58 per cent are of the automatic type of one kind or another; the remainder, 18,580, are hand feed or nonautomatic. There are about 1,856,000 woolen spindles reported, of which it is estimated that about 78,000 are frame spindles and the remainder are on mules. As stated before, the worsted spindles include flyers, caps, and rings, but no information is available as to the division and all are included under one heading.

The twenty-year period being considered covers two important decades, the first of which was the rehabilitation period after the War, with its tremendous demands for textiles and other commodities. The end of this period saw the greatest boom in securities in our history, culminating in the crash of 1929. The second decade covers the worst depression this country has ever experienced, which has carried all industry, including the textile, to new lows in percentage of activity. Although the crash of 1929 marks the beginning of the decline in industry as a whole, the textile industry reached its peak back in 1923. From 1923 to 1929 there was a gradual decrease in activity and also in the amount of machinery in place, followed by a sharp decline during the depression and up to the present time.

A good picture of what has happened to the woolen-and-worsted industry can be obtained by plotting the data collected by the Department of Commerce and showing graphically the trend of machinery in place and that active. While this information is probably not 100 per cent complete or correct, and only recently has a distinction been made between automatic and nonautomatic looms, it is the best information available and has been used as the basis for these graphs.

Fig. 1 shows the woolen spindles in place from 1919 to and including July, 1938. The peak of spindles in place was reached in 1926, with a total of about 2,477,000 spindles, and since then there has been a continuous reduction year by year to a total of about 1,856,000 in July, 1938, which is a reduction of 621,000 spindles, or 25 per cent. This averaged about 52,000 spindles a year, or approximately 10 ten-set average mills per year. Viewed from this angle we get some real conception of the extent of this decline.

On the same graph is shown the number of active woolen spindles in any month. A spindle is counted as active if it has been operated at any time during the month. This gives a good graphic picture of the tremendous fluctuations in production in this industry. There is one significant thing to be noted, that, even in the boom years of 1919 and 1923, there were always 200,000 spindles idle, or about 8 per cent. Because of its many ramifications, the woolen industry never can be 100 per cent employed, but there are indications that at that time there was an excess of machinery. While the spindle activity started to drop off in 1923, the number of spindles installed increased until 1926. This graph shows that the beginning of the recession for the woolen industry was about 1923, with a gradual sliding off until 1929, and then the sharp drop with little let up, into the depths of the 1932 depression. The effect of the National Recovery Administration is shown by the burst of activity in 1933, followed by the gradual recovery up to 1935 and the see-saw activity of 1936 and early 1937. Then came another sharp decline which did not stop for practically a year, and out of which the industry is now laboriously fumbling its way.

Fig. 2 gives a similar picture of the worsted spindles in the

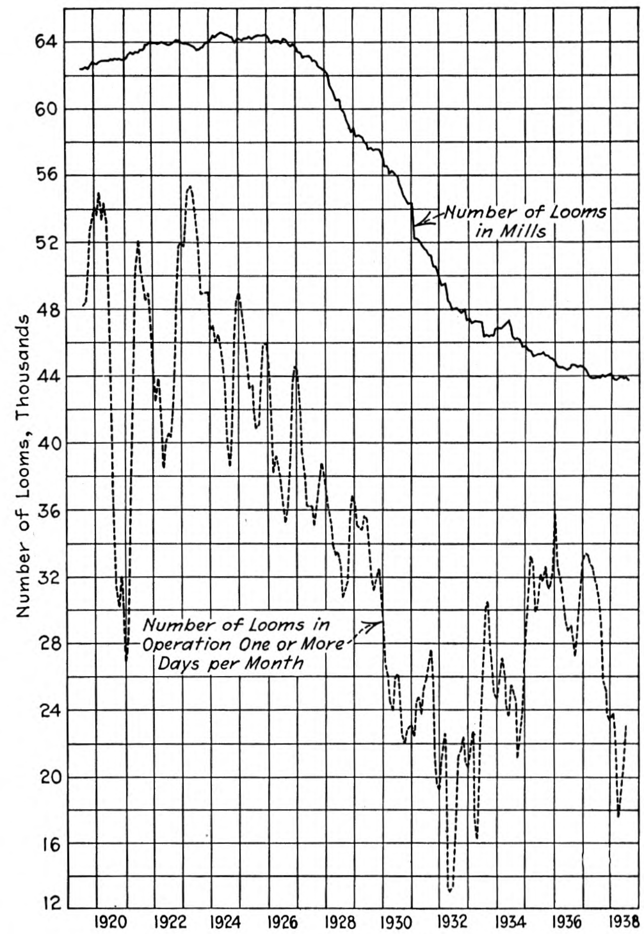


Fig. 3 Woolen and Worsted Broad Looms[2]
(Includes idle mills.)

country. It is of interest that in 1919 there were about 2,400,000 woolen spindles in the country and about 2,350,000 worsted spindles, or practically the same number.

The graph shows that the number of worsted spindles increased to about 2,800,000 in 1924, after which there was a steady decline, and in 1938 the number had been reduced to a little less than 2,300,000, or a decrease of 500,000. This is about 18 per cent as compared with the 25 per cent reduction in woolen spindles. The activity curve is, in general, the same as that of the woolen spindles, except that it is considerably more erratic and the fluctuations are more violent. The year 1923 was the peak point of activity, followed by several years of erratic but always decreasing business, which finally reached a record-low-activity period in 1932. The hypodermic effect of the N.R.A. in 1933 is evident, with its sad aftermath. The partial recovery during 1935 and 1936, as in the woolen industry, was followed by the sharp recession beginning in the early part of 1937, which did not turn upward until the spring of 1938.

It is to be noted that the industry continued to install spindles after the peak of activity had been reached. The fluctuations in machinery activity are so great in the industry that it is difficult at any time to forecast future trends. In 1923 the industry came very close to running all of its spindles for a short time, but since then there has always been a half million or more idle.

This picture would not be complete without showing a graph of the looms in place and their activity. Since the same looms may use both woolen and worsted yarns, they cannot be sepa-

[2] Looms with more than 50 in. reed space are classified as broad looms.

rated as can the spindles, and so they have been grouped into broad looms[2] and narrow looms. The broad looms represent 85 per cent of the total number in place and the narrow looms are of only minor importance.

Fig. 3 shows the number of broad looms in place from 1919 to July, 1938, inclusive. The maximum number of looms was about 64,600, reached in 1924, following the great activity of 1923. There was no appreciable reduction in the number of looms in place until 1926, from which time, as the graph shows, the liquidation has been continuous and rapid until at the present time there are less than 45,000 in place. This is a decrease of 32 per cent or a reduction of about 1700 looms a year. The year of maximum liquidation was 1931, when there was a reduction of about 4600 looms.

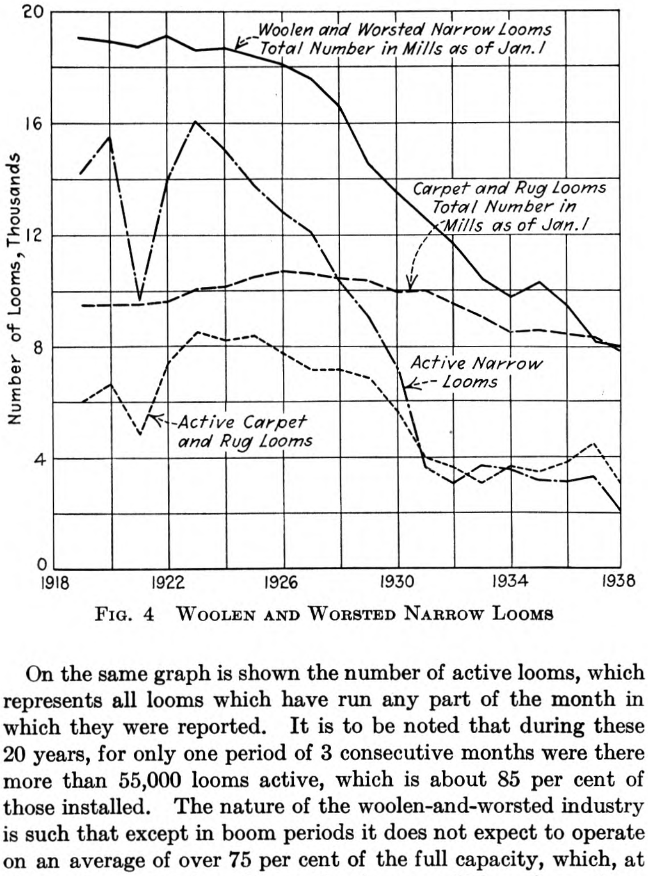


FIG. 4 WOOLEN AND WORSTED NARROW LOOMS

On the same graph is shown the number of active looms, which represents all looms which have run any part of the month in which they were reported. It is to be noted that during these 20 years, for only one period of 3 consecutive months were there more than 55,000 looms active, which is about 85 per cent of those installed. The nature of the woolen-and-worsted industry is such that except in boom periods it does not expect to operate on an average of over 75 per cent of the full capacity, which, at the maximum number of looms in place, would have been about 48,500 looms. Until 1926 the average of activity was in this neighborhood, but in 1930, with 56,000 looms in place, the activity was 25,000; in 1932, the low point, with 48,000 in place, the activity was from 13,000 to 22,000; in 1934, with 46,000 looms in place, the activity was 26,000; and in 1935, with 45,000 in place, the activity was 32,000, or 71 per cent (nearly normal); and in 1938, with 44,000 looms in place, the activity is 22,000, or 50 per cent.

This graph definitely shows a sharp division between the first and second decade of the period considered. During the ten years previous to 1928, the activity was seldom less than 36,000, while in the years since then it has never reached 36,000.

In order to complete the presentation of the loom situation, Fig. 4 has been prepared to show narrow looms in place and active. This shows that there were 19,000 narrow looms in place in 1919 and that in 1938 only 8000 of them remained. It also shows that the peak of activity was in 1923 with 16,000 looms running, and that at the present time there are only 2000 active. It is to be observed that the narrow-loom activity was between 3000 and 4000 looms from 1930 to 1937. The demand for narrow goods has been decreasing steadily and this is reflected in the loom activity. It is not possible to say just how far this will go, but it would seem from the indications that it may reach the point of almost complete extinction. It would naturally be assumed that the broad-loom activity would have been benefited by the decrease in narrow fabrics, and undoubtedly it was, but in spite of this the number of broad looms decreased steadily during this period.

On this graph also is shown the carpet looms, in place and active, in the United States. These are shown only as a matter of passing interest.

The graphs which have been presented show the extent of the liquidation of the spindles and looms in the industry during the 20-year period, and the decrease in activity which has been even more marked. The question naturally rises as to the causes for this apparent wiping out of from 25 to 50 per cent of an industry in 20 years.

A number of things may have contributed to this result, such as, (1) more efficient machinery, (2) lower consumption of woolen and worsted fabrics, and (3) a greater number of hours of operation of the active machinery.

There have been some rather radical changes in a few operations which would have had some effect on the amount of machinery required.

The spinning of woolen yarns on frames instead of mules has reduced the number of spindles required. The best information available indicates that there are about 78,000 frame spindles which, under average conditions, it is estimated would have replaced about 120,000 mule spindles, so that there might have been expected a reduction of 42,000 spindles due to this factor alone, but there was a reduction of 600,000.

The introduction of ring spinning in place of cap spinning in the making of worsted yarns might account for some reduction in this machinery, but not for the 500,000 spindles which have disappeared.

The wide adoption of automatic looms was a real factor, however, in the number of looms required. Twenty years ago probably not more than 3000 looms, or about 5 per cent were automatic. Today, there are reported to be 26,307 automatic looms, many of them of the latest models operating at much higher speeds and efficiency than the old nonautomatic looms. Some measure of this can be obtained by a study of the ratio of spindles to looms for the 20-year period. Table 2, giving the numbers of spindles and looms every 5 years, shows the trend.

TABLE 2 RATIO OF SPINDLES TO WOOLEN AND WORSTED BROAD LOOMS

(Machinery in place)

Year	Number of spindles in thousands: Woolen	Worsted	Total	No. of looms	Spindles per loom
1919	2393	2354	4747	62289	76.0
1924	2445	2718	5163	64111	80.5
1929	2345	2638	4983	58363	85.6
1934	1994	2368	4362	46834	93.0
1938	1866	2281	4147	44887	92.2

The average number of spindles per loom has increased from 76 to 93, or about 22 per cent, in spite of the fact that the spindle production per hour has also increased. Although there are no data available, it is quite certain that where the fabrics will permit, the automatic looms are run in preference to hand-feed looms. It is probably a conservative statement to say that the average active loom-hour today produces 20 per cent more than the loom-hour of 1923. This would account for a reduction of about 13,000 looms, whereas the actual reduction has been about 21,000.

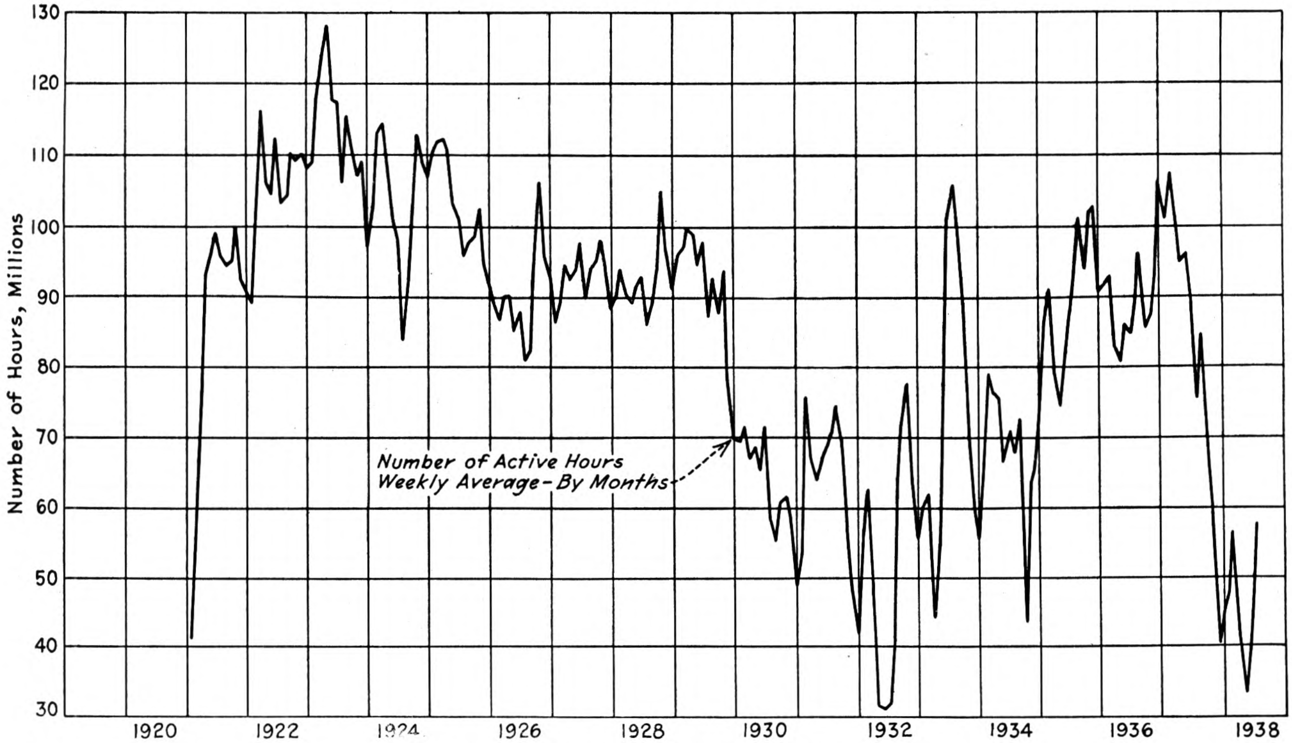


FIG. 6 ACTIVITY OF WOOLEN-SPINNING SPINDLES
(Active hours, weekly average by months.)

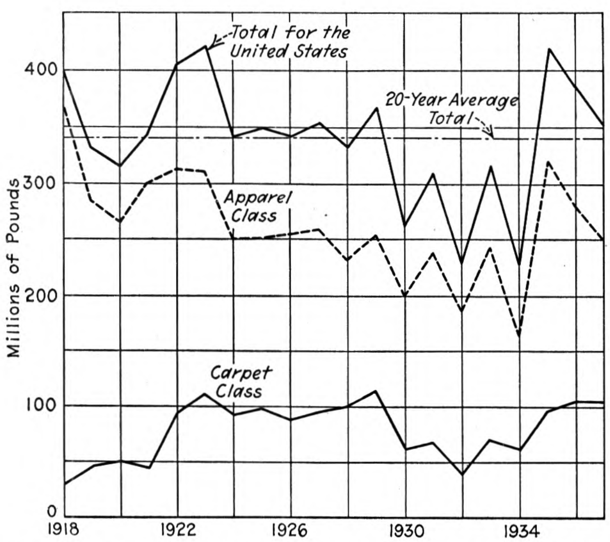


FIG. 5 RAW-WOOL CONSUMPTION BY CLASS, SCOURED BASIS

It would seem, therefore, that machinery changes alone account for only a part of the liquidation.

The question of consumption changes presents some difficulties. The worsted-and-woolen industry consumes a great variety of raw materials besides wool, such as animal hair, cotton, rayon, and silk, and in addition uses large quantities of reworked wools.

The information available from the government reports does not cover all years, and is in itself a little confusing. However, Table 3 shows the various materials used per year for the years when it was reported. The figures are probably not entirely correct for the minor items, but are reasonably so for the main ones and any inaccuracy would not affect the results materially.

The totals would indicate that there has been no general downward trend in production. The years from 1925 to 1929 were normal years and consumption was practically constant. The low production of 1931 is accounted for by the depression, and the 1935 peak by the increasing production from the 1932 low point. The average for the years reported is 523.45 millions of pounds of materials used, which is close to the more normal years from 1925 to 1929.

In this same table is given the total time which the looms were run for each year. From this was determined, at least arithmetically, the average production in pounds per loom per hour, which is given in the table. It probably is not entirely correct but it indicates that the production per loom-hour does not vary too much even with wide fluctuations in total production, and that the trend is toward higher levels, as one would expect.

Since the principal raw material of the woolen-and-worsted industry is wool, Fig. 5 was prepared, which shows the total consumption of scoured wool in the United States per year and that used by the apparel industry.

While the total consumption during the last 10 years is less than in the 10 preceding years, the 1935 peak would indicate that, under more normal conditions, consumption of wool might

TABLE 3 PRINCIPAL MATERIALS USED IN WOOLEN-AND-WORSTED-GOODS INDUSTRY

	Millions of pounds used in					
Material	1935	1931	1929	1927	1925	1919
Wool, scoured basis...	417.6	311.0	368.1	354.1	349.9	329.1
Animal hair........	27.3	15.4	27.5	26.1	27.2	28.3
Cotton............	12.5	14.5	20.2	22.8	26.6	17.3
Rags, clips, and recovered wool......	111.4	51.8	93.0	81.6	106.5	79.6
Rayon............	12.7	2.6	3.5	1.5	2.5	....
Silk and spun silk....	0.65	0.45	1.2	1.9	0.6	0.4
Cotton yarns.......	37.3	27.0	43.0	31.5	26.8	28.7
Total materials....	619.45	421.75	556.50	519.50	540.10	483.40
Yearly run, millions of loom-hours.....	95.6	74.4	95.6	98.6	112.8	....
Pounds per loom-hour, average.....	6.50	5.66	5.82	5.23	4.78	....

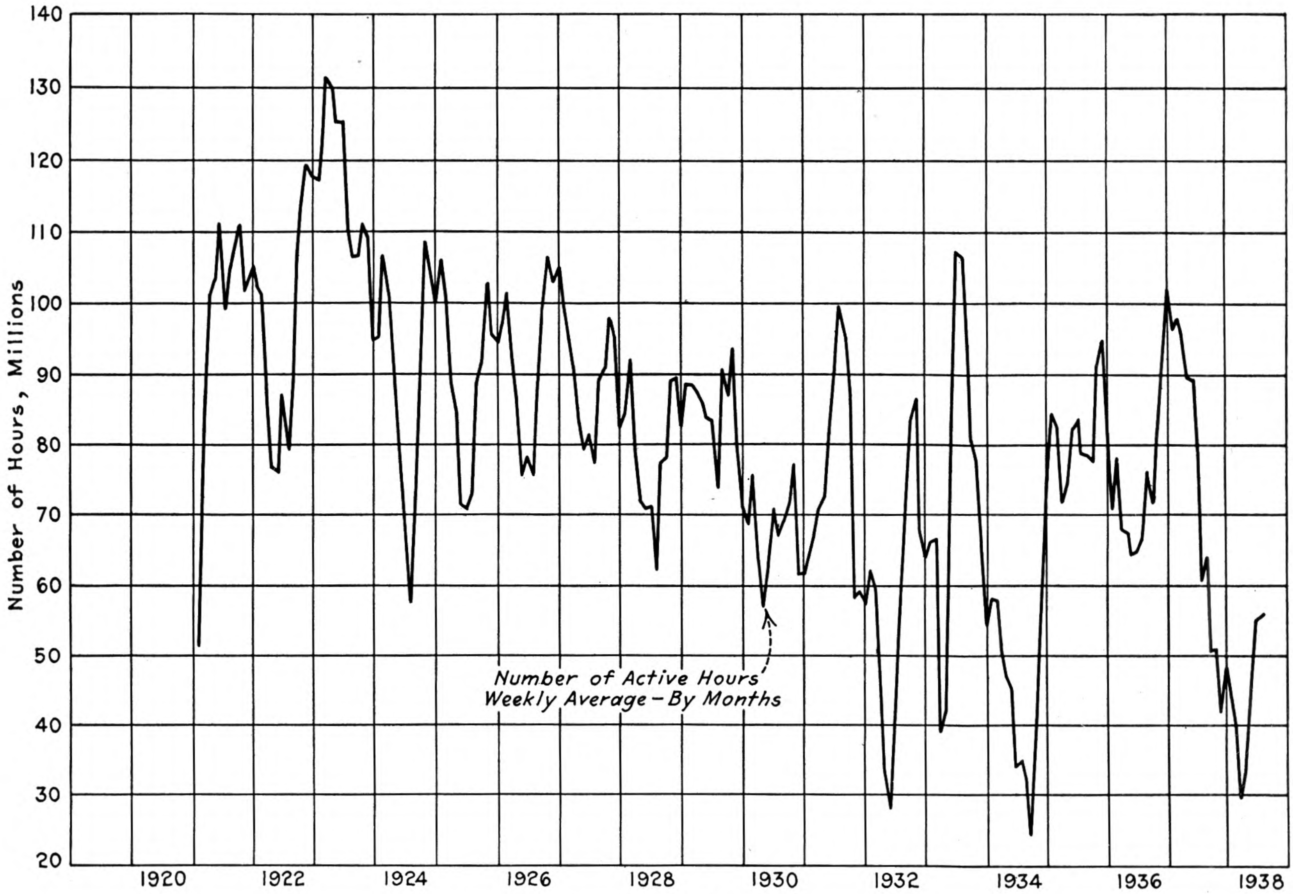


FIG. 7 ACTIVITY OF WORSTED-SPINNING SPINDLES
(Active hours, weekly average by months.)

be expected to be as much, if not more, than was consumed from 1919 to 1929. The table previously given, of all materials consumed, would lead to the same conclusion.

It is a logical conclusion that the use of higher production and more efficient machinery does not account for the decrease in machinery installed, and also that there is no permanent reduction in the quantity of raw materials consumed. In 1935, 319,000,000 lb of wool were consumed for apparel purposes as compared with 312,800,000 lb in 1922, with 534,000 fewer spindles, and 15,700 fewer looms.

And so the next point is the consideration of the hours of operation of the machinery in the industry. The actual number of spindle- or loom-hours operated per week or per year is, after all, the best measure of the condition of the industry, and a good indicator of consumption. It will be recalled that the amount produced per loom-hour varies little between periods of high and low consumption. There have been prepared some graphs of spindle- and loom-hours operated, which are interesting. These graphs show by months the weekly average for the month of the hours of operation. They do not show the maximum and minimum weeks but these would not be very different.

Fig. 6 shows the woolen-spindle activity on this basis. The graph accentuates what has been indicated on some of the previous graphs, and that is the higher general average of production of the first decade over the second. For the first time a real picture is given of the activity of recent years as compared with the period prior to 1930. In 1933, 1935, and 1937, the hours of operation reached a plane which compares favorably with the heights reached prior to 1930.

Fig. 7 shows the same information for the worsted spindles. It is to be noted that the annual fluctuations for these spindles are much greater than for woolen spindles. Peaks of activity and the depths of slack periods are more extreme than in the woolen industry but the general shape and trend are the same in both cases. It is the general impression that operations are much more erratic now than previous to 1929 and this graph would seem to bear out that impression, but it is also evident that the earlier period was not without its troubles. The greatest slump was the one that started in the middle of 1933 and continued downward for 15 months, almost without a stop.

Fig. 8 shows the average weekly number of hours of broadloom operation, plotted by months. In general form, it follows the two previous graphs with its peaks and depressions coinciding as to time and varying as much in their swings.

The loom-hour is probably the best index of production in the woolen-and-worsted industry and because many looms use both woolen and worsted yarns, its activity is a reflection of the conditions in both branches. There is a noticeably general downward trend from 1921 to 1932 and what appears to be an upward trend since that time. The 1937 recession cast some doubt as to "where we go from here." It must be remembered that, in the period from 1926 to 1938, the number of looms in place dropped from about 64,600 to about 43,900 in spite of the fact that the 1937 consumption of raw stock was the maximum since 1919 when the keeping of such records was started.

Another interesting observation is that it required 12,750,000 loom-hours to consume 540,100,000 lb of raw stock in 1925, and in 1935 only 10,400,000 loom-hours for 619,450,000 lb of raw stock.

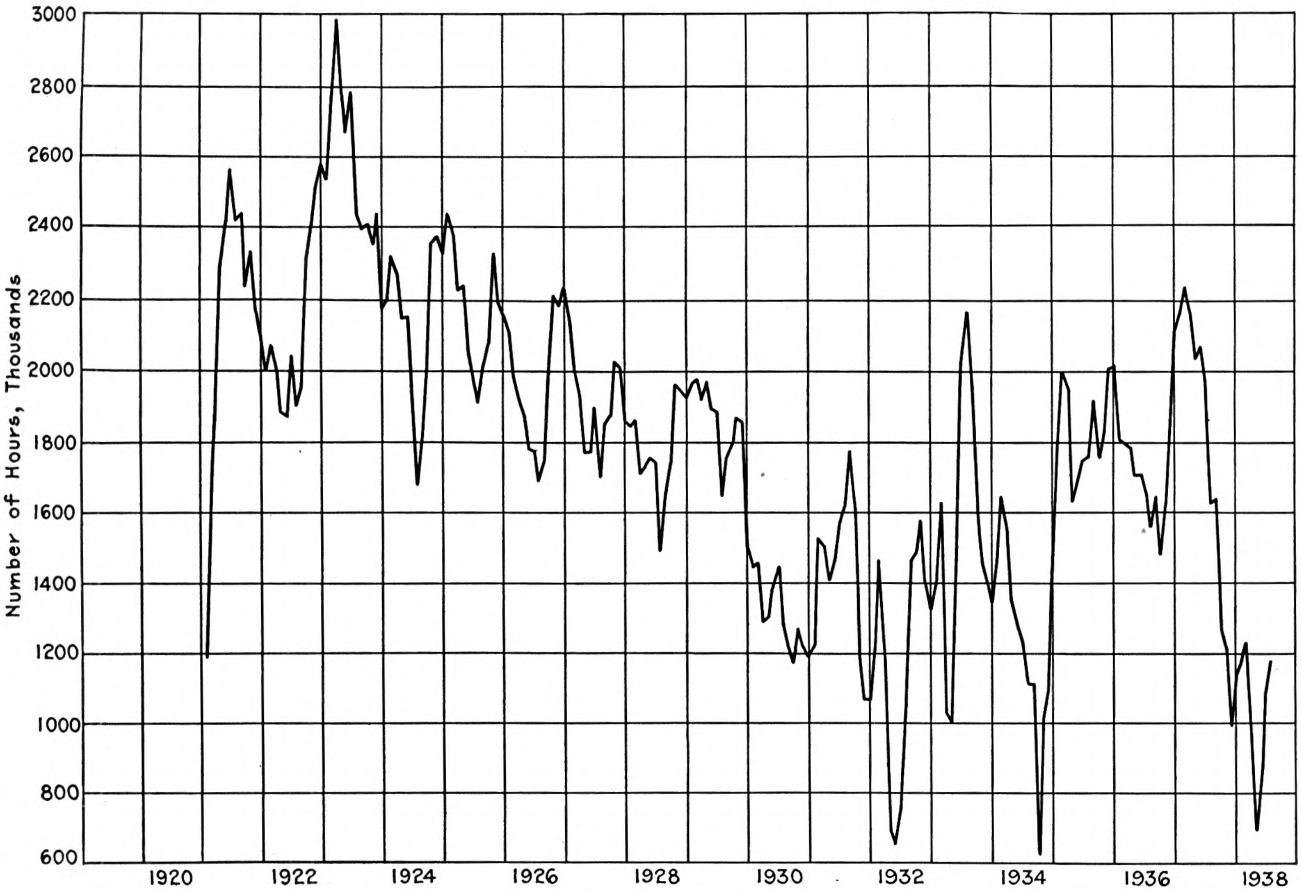


FIG. 8 ACTIVITY OF WOOLEN AND WORSTED BROAD LOOMS
(Active hours, weekly average by months.)

In order to properly evaluate such comparisons, there are tabulated in Table 4 several computations which were made of woolen and worsted spindles and looms at the beginning and end of the 20-year period. The raw-consumption figures are only available for certain years and the hours of operation only since 1921, so the years 1925 and 1935 were chosen because complete information was available for those years. However, since it covers only ten years, Table 4 does not reflect the full change in the 20 years.

TABLE 4 COMPARISON OF HOURS OF OPERATION IN 1925 AND 1935

WOOLEN

Year	Thousands of spindles In place	Thousands of spindles Active	Approximate average, spindle-hours per week	Hours run per week Spindles in place	Hours run per week Active spindles
1925	2420	1950	92,700,000	38.4	48
1935	1950	1550	90,500,000	46.5	58.5

WORSTED

Year	Thousands of spindles In place	Thousands of spindles Active	Approximate average, spindle-hours per week	Hours run per week Spindles in place	Hours run per week Active spindles
1925	2760	1900	89,000,000	32.5	47
1935	2340	1650	82,500,000	35.3	50

LOOMS

Year	Looms In place	Looms Active	Approx. average loom-hours per week	Hours run per week Looms in place	Hours run per week Active looms	Raw materials, millions lb
1925	64200	44200	2,168,000	34	49	540.1
1935	45200	32070	1,847,000	41	57.5	619.4

There are certain observations which can be drawn from Table 4, as follows:

1 While the number of woolen spindles in place was reduced 23.5 per cent, and the number of active spindles 20 per cent, the average weekly number of spindle-hours increased 21 per cent for those in place and 22 per cent for those active.

2 The number of worsted spindles decreased by 15 per cent of those in place and 13 per cent of the active spindles, but the number of hours operated per week for both in-place and active spindles increased only 6 per cent.

3 From the total yearly number of spindle-hours run by woolen and worsted spindles in 1925 and the raw-stock consumption of 540 million pounds per year, it is found that each spindle accounted for 2.70 lb of stock per week, whereas the corresponding figures for 1935 show that each spindle accounted for 3.76 lb per week, or an increase of 39 per cent.

4 The raw-stock consumption of 1925 divided by the loom-hours operated per year, gives 234.7 lb of raw-stock required to supply an active loom for an average week, while in 1935 an active loom, operating for its average weekly number of hours required 372.9 lb of raw stock, or an increase of 59 per cent.

The year, 1925, was quite normal for that period as to raw-stock consumption and hours of operation. The year, 1935, was a year of high consumption and a quite different setup as to hours of operation. In 1925 the mills in general operated one shift with some overtime, but in 1935 they were endeavoring to run two 40-hr shifts per week.

The figures would indicate that the production per loom had increased materially between 1925 and 1935. Eventually the 1937 figures will be available and this trend can be traced further.

The industry, because of its many ramifications in stocks and fabrics, the wide fluctuation in seasonal activity, and the relatively narrow range of yarns over which spindles can operate, is bound to have a considerable number of spindles idle at any one time even though they all operate on an average 75 per cent of the year. It is not clear, therefore, that there will be much more liquidation of spindles except that due to the introduction of more efficient machinery.

The situation with reference to looms is somewhat different. Table 5 shows the average weekly number of hours operated per loom from 1921 to 1937. The first column is the weekly average for the year; the second column is the weekly average for the maximum month and the third column is the annual consumption of raw stock for the years in which it is available.

TABLE 5 ACTIVITY OF WOOLEN AND WORSTED BROAD LOOMS

Year	Active loom-hours, in thousands: Average week of year	Active loom-hours, in thousands: Average week of maximum month	Raw stock consumed, millions lb
1921	2140	2563	...
1922	2130	2577	...
1923	2560	2986	...
1924	2140	2389	...
1925	2170	2433	540
1926	1960	2235	...
1927	1900	2136	520
1928	1780	1963	...
1929	1840	1984	557
1930	1310	1463	...
1931	1430	1783	422
1932	1190	1584	...
1933	1550	2167	...
1934	1250	1649	...
1935	1840	2021	619
1936	1760	2119	...
1937	1710	2244	...
17 years' average.....	1810	2140	...

The noticeable feature is the steady decrease in loom-hours over the period without a corresponding decrease in raw-stock consumption. Assuming that the consumption of woolen and worsted fabrics is to continue at about the same rate as the present average year, which is reasonable, the figures would indicate that the average number of loom-hours required per week will be around 1,800,000 and the maximum not over 2,250,000. This gives a measure which can be applied to determine the number of looms which the industry can support for different numbers of weekly operating hours. Table 6 shows the results of applying this yardstick.

TABLE 6

Hours operated per week	No. of looms[a]: For 100% operation	No. of looms[a]: For 75% operation	No. of looms[b]: For 100% operation	No. of looms[b]: For 85% operation
40	45250	60400	56100	66000
54	33500	45000	41600	49000
80	22600	30000	28100	33000
108	16800	22400	20900	24600
120	15100	20100	18700	22000

[a] Based on an average number of 1,800,000 loom-hours per week.
[b] Based on an average number of 2,250,000 loom-hours per week.

On the basis of 1,800,000 loom-hours average per week, column 2 would indicate the number of active looms and column 3 the looms in place assuming 75 per cent activity for the conditions indicated in column 1. Column 4 gives the number of active looms on the basis of using 2,250,000 loom-hours per week average but it is assumed that if the activity reached this high level there would be only 15 per cent idle and column 5 gives the number of looms in place using this assumption.

It is probably useless to try to predict the actual hours of operation which legislation or other agencies will determine, but it is highly improbable that the work week will be a single 40-hour shift, because of insufficient capacity, or a single 54-hour shift because of opposition from many sources. Neither is the third shift likely to be acceptable in the long run, and operation for 120 hours per week is therefore doubtful. The most probable hours will be 80 per week, or possibly a few more. From the table it is seen that this would not require more than 33,000 looms, or about 10,000 fewer than are now in place. No prediction to this effect can be made, but a study of the graph of looms in place gives no clear indication that the liquidation has ceased. It is possible that there is no great excess of spindles but the further introduction of automatic looms must necessarily reduce the number required and the new automatic looms which can be converted to 4 × 4 box looms for pick-and-pick fabrics eliminates the necessity of keeping the older looms for that purpose, and again reduces the necessary number of looms in place.

The author is inclined to believe that the liquidation of woolen and worsted mills as a whole is nearing its close although there will be some further reduction in numbers of spindles and looms in operating plants.

The loom situation, however, is not quite as bright and there are many indications that the number of looms can be further reduced without any lessening of the amount of goods produced. This, again, does not mean the liquidation of existing weaving and finishing plants, but rather a reduction in the number of looms which they will operate.

The making of woolens and worsteds is a basic industry and its continuance at its present average rate of production is an absolute necessity. This does not insure, however, that because a plant is in existence today it must continue; only those who keep abreast of the times in their mechanical equipment, who foresee style trends, and who exercise good judgment in the management of their plants can expect to go on.

The Elastic Theory of Wood Failure

By CHARLES B. NORRIS,[1] GRAND RAPIDS, MICH.

This paper deals with various theories advanced for the failure of wood. The formulas developed by Jacoby, Howe, and Hankinson for expressing the crushing strength of wood are examined and their relative practical value indicated. Of the three the Hankinson formula, while being strictly empirical, agrees with experimental results better than the others. The author then proceeds to outline the steps of derivation of Hankinson's formula, applying the theory of failure advanced by Hencky. A second theory is analyzed which also leads to Hankinson's formula. The author points out that the methods used in deriving Hankinson's formula are very approximate but that such a formula is better than none.

The FAILURE of wood has long been studied by the Forest Products Laboratory. Publication has been made of the strength values of woods grown in the United States when the stress is applied either parallel or perpendicular to the grain (1).[2] This study was greatly accelerated during the World War due to the extensive use of wood in the construction of aircraft. Numerous studies have also been made of the compressive strength of wood when the stress is applied at an angle to the grain of the wood. As a result various formulas have been developed, notably those of H. S. Jacoby, N. A. Howe, and R. L. Hankinson (2).

These formulas all express the crushing strength of wood N, in terms of the crushing strength of the wood parallel to the grain P, the crushing strength of the wood perpendicular to the grain Q, and the angle to the grain θ at which the stress is applied. The formulas are

Jacoby.............. $N = P\cos^2\theta + Q\sin^2\theta$

Howe............... $N = Q + (P - Q)\left(1 - \dfrac{\theta}{90}\right)^{5/2}$

Hankinson........... $N = \dfrac{PQ}{P\sin^2\theta + Q\cos^2\theta}$

The Jacoby formula gives values which are too high compared with experimental results. The Howe and Hankinson formulas give very similar curves, especially for values of θ over 25 deg. The Hankinson formula is strictly empirical and agrees with the experimental results better than the other two. It seems to be the best formula but it cannot be applied to more general conditions of stress because of its empirical nature.

Considerable light is thrown upon the problem by applying the theory of failure developed by H. Hencky (3). This theory is very clearly described by A. Nádai (4). In this theory the total energy per unit volume, stored in a body due to elastic distortion, is divided into two parts, that due to the change in volume and that due to the change in shape. It is assumed that, when the elastic energy due to the change in shape reaches a certain value, the material changes from an elastic condition to a plastic one, i.e., it starts to fail. This has been found to be true of a great number of materials. The energy constant is different for each material but is independent of the manner in which the material is stressed.

The total energy stored in the material per unit volume is

$$\tfrac{1}{2}(s_1e_1 + s_2e_2 + s_3e_3) \qquad [1]$$

The energy stored due to a change in volume is

$$\tfrac{1}{6}(s_1 + s_2 + s_3)(e_1 + e_2 + e_3) \qquad [2]$$

where s_1, s_2, s_3 are the principal stresses and e_1, e_2, e_3 are the principal strains.

Subtracting Equation [2] from [1], the energy stored per unit volume, due to the change of shape, is obtained as follows

$$\tfrac{1}{6}(2s_1e_1 + 2s_2e_2 + 2s_3e_3 - s_1e_2 - s_1e_3 - s_2e_1 - s_2e_3 - s_3e_1 - s_3e_2) \qquad [3]$$

For simple compression $s_2 = s_3 = 0$, and Equation [3] becomes

$$\tfrac{1}{6}(2s_1e_1 - s_1e_2 - s_1e_3) \qquad [4]$$

or

$$\frac{s_1}{6}(2e_1 - e_2 - e_3) \qquad [5]$$

The three relations are

$$e_1 = \frac{s_1}{E_1}, \qquad e_2 = -\frac{s_1}{E_1}r_1, \qquad e_3 = -\frac{s_1}{E_1}r_2$$

where, r_1 and r_2 are the Poisson ratios in the two directions and E_1 is the modulus of elasticity in the direction of the stress s_1. Substituting these values in Equation [5], the energy per unit volume due to the change in shape becomes

$$h = \frac{s_1^2}{6E_1}(2 + r_1 + r_2) \qquad [6]$$

If this equation is applied to wood, first applying the stress in a direction parallel to the grain and then in a direction radial to the annular rings, the two following equations are obtained

$$h_L = \frac{s_L^2}{6E_L}(2 + r_{LR} + r_{LT}) \qquad [7]$$

$$h_R = \frac{s_R^2}{6E_R}(2 + r_{RL} + r_{RT}) \qquad [8]$$

At the point of failure

$$H = \frac{P^2}{6E_L}(2 + r_{LA} + r_{LR}) \qquad [9]$$

and

$$H = \frac{Q^2}{6E_R}(2 + r_{RL} + r_{RT}) \qquad [10]$$

Substituting values obtained from Equations [9] and [10] in

[1] Consulting Engineer, Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Presented at Joint Meeting of the Wood Industries Division of The American Society of Mechanical Engineers and the Cost and Production Division of the Southern Furniture Manufacturers' Association at High Point, N. C., September 21–23, 1938.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until May 10, 1939, for publication at a later date.

Equations [7] and [8], the following values of the energy stored per unit volume due to the change of shape are obtained

$$h_L = s_L{}^2 \frac{H}{P^2} \dots\dots\dots\dots\dots\dots [11]$$

$$h_R = s_R{}^2 \frac{H}{Q^2} \dots\dots\dots\dots\dots\dots [12]$$

The first equation applies when the stress is applied in a direction parallel to the grain and the second when the stress is applied in a direction radial to the annular rings. The value of H is the same in both equations because the theory assumes that failure will take place at the same energy in either case.

Now, if a stress s is applied to a block of wood at an angle g to the grain, the state of stress of the wood can be described in terms of a stress parallel to the grain, a stress perpendicular to the grain, and a shear stress. These relations may be expressed as

$$s_L = s \cos^2 g \dots\dots\dots\dots\dots\dots [13]$$

$$s_R = s \sin^2 g \dots\dots\dots\dots\dots\dots [14]$$

$$q = s \sin g \cos g \dots\dots\dots\dots\dots\dots [15]$$

The values given by the first two of these equations may be substituted in Equations [11] and [12] to obtain that part of the elastic energy due to each of the stresses s_L and s_R.
Thus

$$h_L = s^2 \frac{H}{P^2} \cos^4 g \dots\dots\dots\dots\dots\dots [16]$$

$$h_R = s^2 \frac{H}{Q^2} \sin^4 g \dots\dots\dots\dots\dots\dots [17]$$

The energy caused by the shear stress is all due to a change of shape. This energy is given by

$$h_S = \frac{s^2}{N} \sin^2 g \cos^2 g \dots\dots\dots\dots\dots\dots [18]$$

where N is the rotational modulus of elasticity.

The sum of these three last equations gives the energy per unit volume due to the change of shape of the block of wood. This sum is equal to H at the point of failure.

$$H = s^2 \left[\frac{H}{Q^2} \sin^4 g + \frac{1}{N} \sin^2 g \cos^2 g + \frac{H}{P^2} \cos^4 g \right] \dots [19]$$

It is desired to find the value of H which will reduce this equation to Hankinson's formula. If

$$H = \frac{2PQ}{N} \dots\dots\dots\dots\dots\dots [20]$$

this condition is satisfied. Both sides of [19] can be divided by H and the expression in the bracket becomes a perfect square. Taking the square root of both sides of Equation [19]

$$1 = s \left[\frac{1}{Q} \sin^2 g + \frac{1}{P} \cos^2 g \right]$$

Multiplying both sides by PQ and solving for s

$$s = \frac{PQ}{P \sin^2 g + Q \cos^2 g}$$

The foregoing derivation of Hankinson's formula is of course an approximate one. Yet by it Hencky's theory is made to apply to wood. The relation expressed by Equation [20] is a fundamental one. This relation can be used and the theory applied to more complicated conditions of stress. The theory should be checked by tests in which two separate stresses are applied to a block of wood at the same time.

The value of H can also be obtained by the use of both Equations [9] and [10] if the values of Poisson's ratios are known. The two values obtained in this way do not agree very well. This fact might be expected since the measured values of Poisson's ratios are only virtual because of the cellular structure of wood. The energy value given by Equation [20] should be the true elastic energy due to the change of shape of the cell walls.

Another theory of failure of wood can be devised which does check very well with the virtual values of Poisson ratios determined by test. This second theory also leads to Hankinson's formula and may be equivalent to the theory just given, the difference being due to the use of the true Poisson's ratios in one case and the virtual values in the other.

The criterion of failure in this theory is that the expression $(2e_1 - e_2 - e_3)$ reaches a certain value. In examining the meaning of this term, consider a block of material subjected to stress. The material is given the unit deformations e_1, e_2, and e_3 by the application of the stress. Of these e_1 is the greatest. Assume now that stresses are added so that these three deformations become equal. The expression above becomes zero. If the extra stresses are relieved, a unit volume of the material will change. The expression $(2e_1 - e_2 - e_3)$ is exactly equal to that change in volume. The units in which it is measured are cubic inches per cubic inch. According to this, as well as to Hencky's theory, hydrostatic pressure is incapable of causing failure. Of course this is not true of a cellular structure like wood, unless the liquid enters the spaces within the cells.

The development of this theory is similar to the development of the first theory but much simpler. The change in volume, when the extra stresses are removed, is of course given by

$$k = 2e_1 - e_2 - e_3 \dots\dots\dots\dots\dots\dots [5a]$$

If the three stress-strain relations given in the first part of this paper are applied to this equation it becomes

$$k = \frac{s_1}{E_1} (2 + r_1 + r_2) \dots\dots\dots\dots\dots\dots [6a]$$

The method is parallel to that just given.

Then

$$K = \frac{P}{E_L} (2 + r_{LR} + r_{LT}) \dots\dots\dots\dots [9a]$$

$$K = \frac{Q}{E_R} (2 + r_{RL} + r_{RT}) \dots\dots\dots\dots [10a]$$

$$k_L = s_L \frac{K}{P} \dots\dots\dots\dots\dots\dots [11a]$$

$$k_R = s_R \frac{K}{Q} \dots\dots\dots\dots\dots\dots [12a]$$

If the two relations given by Equations [13] and [14] are applied to the last two of these equations then

$$k_L = s \frac{K}{P} \cos^2 g \dots\dots\dots\dots\dots\dots [16a]$$

$$k_P = s \frac{K}{Q} \sin^2 g \dots\dots\dots\dots\dots\dots [17a]$$

The shear strain does not change the volume of the block of wood, so it does not affect the value of K and is therefore omitted.

The total change in volume caused by removing the extra stresses referred to is given by the sum of the foregoing expressions. At the point of failure

$$K = s\left[\frac{K}{Q}\sin^2 g + \frac{K}{P}\cos^2 g\right] \ldots\ldots\ldots\ldots [19a]$$

The volume-change constant K cancels out and it can be seen at once that the equation reduces to Hankinson's formula.

The volume-change constant K can be obtained from either Equation [9a], Equation [10a], or a third one designated as Equation [10b]

$$K = \frac{R}{E_T}(2 + r_{TL} + r_{TR}) \ldots\ldots\ldots\ldots [10b]$$

The agreement between the values obtained by the use of these three equations is very good. The Poisson ratios for various woods are not very well known. However, C. F. Jenkin (5) has given values for spruce, mahogany, ash, and walnut with the other elastic constants and the strength values for the same pieces of wood. The values for spruce under compression will suffice to show the agreement obtained. These values are

$$K = 0.00905, \quad 0.00635, \quad 0.00962$$

Considering the difficulty in obtaining accurate test data the agreement seems very good indeed.

The author realizes that the methods used in deriving Hankinson's formula are very approximate. There are twenty-one elastic constants for a material such as wood (6). Only ten have been used in this work. However an approximate formula is better than none. If this paper stimulates interest in the theory of wood failure it will have served its purpose.

BIBLIOGRAPHY

1 "Strength and Related Properties of Woods Grown in the United States," by L. J. Markwardt and T. R. C. Wilson, Technical Bulletin No. 479, September, 1935, U. S. Department of Agriculture, Washington, D. C.

2 "Wood Handbook," furnished by the Forest Products Laboratory, Madison, Wis., U. S. Department of Agriculture, Washington, D. C., 1935.

3 "Zur Theorie Plastischer Deformationen," by H. Hencky, *Zeitschrift für angewandte Mathematik und Mechanik,* vol. 4, 1924, pp. 323–334.

4 "Plasticity," by A. Nádai, Engineering Societies Monograph, McGraw-Hill Book Company, Inc., New York, N. Y., 1931.

5 "Report on Materials of Construction Used in Aircraft," by C. F. Jenkins, Great Britain Aeronautical Research Committee, 1920, p. 105.

6 "Introduction to Theoretical Physics," by Leigh Page, Second edition, D. Van Nostrand Company, Inc., New York, N. Y., 1935.

Discussion

Energy Distribution in the Pulverized-Coal Furnace[1]

ADDENDA AND CORRECTIONS

W. J. Wohlenberg.[2] While the use of radiant-mean positions of burning and ash particles for the purpose of approximating the equilibrium solution in Parts I to IV of the paper is valid, extension of their use for approximating the distribution of energy absorption as heat over the cold walls is not valid. The latter yields the artificial distribution shown in Fig. 9 of the paper. This could exist only if the particles were actually concentrated in their radiant-mean planes. Hence the secondary peak shown in curve No. 1 is the result of an assumed condition and has no place in reality.

It follows that Part V of the paper from its beginning to the heading "General Deductions and Conclusions" and Appendix 5 each involve such errors of assumption and should be replaced by the information submitted herewith for that purpose. Certain other omissions also are noted later in this discussion.

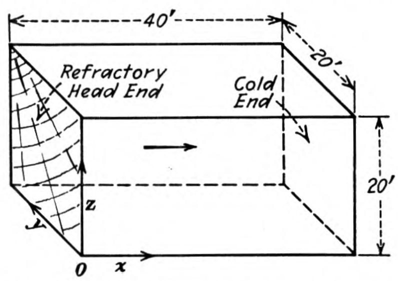


Fig. 1 Conditions in Furnace Cavity

(The cavity is single flow with charge entering through nozzles uniformly distributed over head-end wall. Since the cross section is a square and the longitudinal walls are alike (cold), the origin O may be taken at any one of four corners at the head end and the resulting pattern of contours of energy absorption as heat over cold walls will be the same. The fuel is carbon with 10 per cent ash, pulverized to 70 per cent through 200 mesh, and fired with 120 per cent air at 15,000 Btu per cu ft per hr. Radiant-mean temperatures from equilibrium solution are, in deg F abs: $T_u = 3460$, $T_r = 2540$, $T_g = 2465$, $T_j = 2453$, and $T_c = 1000$.)

Two of the latter affect the equilibrium solution in Parts I to IV. When these corrections are introduced the radiant-mean-temperature values change from those shown in Figs. 8 and 12*b* of the paper to those given here with Fig. 1. The total energy absorbed by the furnace walls as corrected is now 51.4 per cent of that supplied instead of 48.8 per cent as shown in Fig. 8 of the paper. Other details of this figure are of course also affected slightly but not sufficiently to alter either the qualitative nature of the results or the discussion with respect to them.

For the purpose of evaluating the distribution of energy absorption as heat over the cold walls of the cavity from the data obtained in the equilibrium solution it is necessary to consider the burning and ash particles in their real instead of in their radiant-mean position. The net radiating power of burning particles and ash particles is then determined with respect to each point of cold surface which it is proposed to investigate. This of course must also be done for radiation absorption at such points from the radiating gas molecules and the refractory wall. The relations and methods by means of which this is accomplished are stated later.

Referring to Fig. 1 herewith the points investigated in the longitudinal walls of the cavity $z = 0$ were chosen in lines $y =$ const with x variable. In the cold end the points were taken at $z = y$, that is, in lines connecting diagonally opposite corners. The results when plotted, Figs. 2 to 6, then form contours of energy absorption as heat for the longitudinal walls. For the cold end the points falling in the two similar diagonal contours are indicated.

The curves of Fig. 6 represent the final result sought. They are composed of information contained in Figs. 2 to 5 and so a brief discussion of each will aid in an understanding of this final result.

It is noted that in each figure the contour of lowest absorption rate is that for $y = 0$ where two walls intersect. Minimum points fall in corners at $y = 0, x = 0$ and $y = 0$, $x = 40$ ft. The rate of energy absorption rises more or less rapidly across the cavity from $y = 0$ to $y = 10$ for any given value of x and the contour of maximum absorption rate is in all cases that for $y = 10$. Corresponding contours are of course equally spaced, as to y, on either side of the central line $y = 10$.

Peaking of the contours in Fig. 2 is the net result of two effects which oppose each other. One of these is a geometrical influence which tends to increase the absorption rate at the point, from particles in suspension, as this point moves away from the end wall of the cavity. The other is due to the rapidly decreasing extent of radiating surfaces of burning particles as these move away from the head end of the cavity. The peaks occur in a zone close to the radiant-mean position of burning particles because in fact by far the larger fraction of burning-particle surface is concentrated in a region close to this radiant-mean position. This accounts for the close proximity to each other of contours $y = 1.0$ and $y = 10$ for $x > 5$. Contours for $y > 1$ and < 10 of course fall between those for $y = 1.0$ and $y = 10$. It is seen that the error introduced by computing the distribution of radiant-energy absorption over the cold walls from the radiant-mean position is not a large one in so far as burning particles are concerned. This however is quite the reverse for the ash particles represented in Fig. 3.

It is noted in this case that there is no similarity whatever between the shapes of the contours in Fig. 3 and curve No. 2 in Fig. 9 of the paper, which is supposed to represent the same thing. This clearly illustrates the magnitude of the error in distribution which was introduced on the former basis.

The rather abrupt rise from the left, of the curves shown in Fig. 3, is caused by the fact that as the surface point moves from the head end of the cavity it is rapidly passing into regions of greater and greater concentrations of ash particles.

Comparing Figs. 3 and 4 it is noted that the proportional spacing of contours $y = 1$ to $y = 10$ is considerably greater in Fig. 3. This is due to the difference in effect of radiant-mean thickness of the cloud in the two cases. As shown, this effect is much larger for ash particles in suspension than it is for the radiating gas

[1] Published as paper FSP-60-19, by W. J. Wohlenberg and D. E. Wise, in the October, 1938, issue of the A.S.M.E. Transactions, vol. 60, pp. 531–547.

[2] Professor of Mechanical Engineering, Yale University, New Haven, Conn. Mem. A.S.M.E.

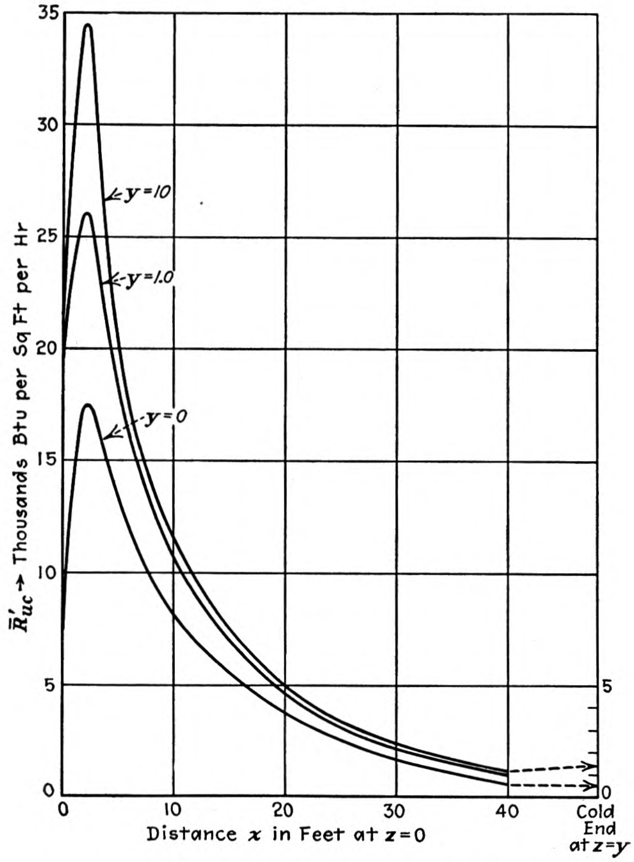


FIG. 2 $\bar{R}'_{uc}$, BURNING PARTICLES ⇆ COLD WALLS
(T_u = 3460 F abs, T_c = 1000 F abs.)

molecules in the range of thickness variations which occur in the present cavity.

In Fig. 5 all contours shown for absorption rate from refractory end, except that for $y = 0$, begin at substantially the same point for $x = 0$. That for $y = 0$ falls in a head-end corner of the cavity. For this corner the angle factor F'_{cr} subtended by refractory walls is 0.25. With $x = 0$ it requires only a small finite displacement from $y = 0$ toward $y > 0$ to increase F'_{cr} from 0.25 to 0.50 at which value it remains for all points $x = 0, 20 > y > 0$. This accounts for the grouping of points shown on the ordinate of Fig. 5, at $x = 0$.

In Fig. 6 any ordinate of the contours of total energy absorption as heat is the sum of ordinates from Figs. 2 to 5 for the given surface point. The absorption rate over the total cold walls as computed on the basis of Fig. 6 differs by 5 per cent from the same thing as computed directly on the basis of the results, found in the equilibrium solution. This difference represents 2.5 per cent of the energy supplied by the fuel. In view of the numerous approximations, some of which are unavoidable, the check is probably as close as should be expected.

With respect to the cold end of the cavity it should be kept in mind that the absorption rates shown are due to radiation only. Hence if the gases leave through this end, the total absorption rate here will be increased considerably because of the large transfer by convection between gases and tubes. This has no influence on the temperature of the contents of the cavity because it does not occur until such contents are leaving the cavity. Thus it is seen how information furnished by the contour lines of Fig. 6 might be employed for estimating the convectional heat transfer at exit more closely.

All of the contours shown in Fig. 6, except that for $y = 1.0$, peak near the left end. Reasons for the peaking are now quite

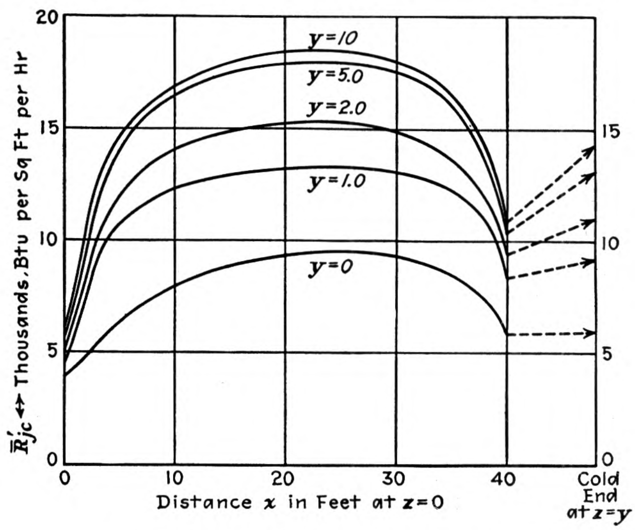


FIG. 3 $\bar{R}'_{jc}$, ASH PARTICLES ⇆ COLD WALLS
(T_j = 2453 F abs, T_c = 1000 F abs.)

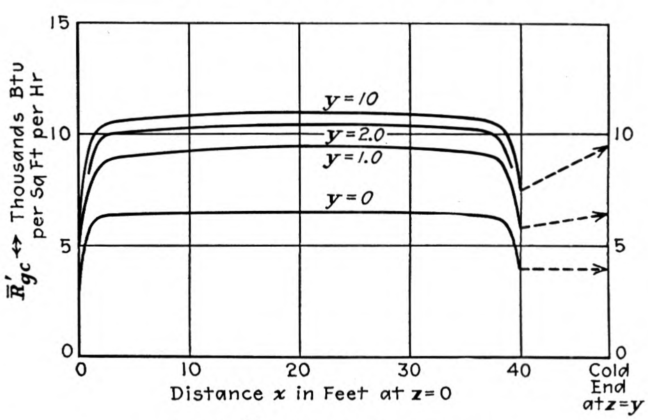


FIG. 4 $\bar{R}'_{gc}$, GAS ⇆ COLD WALLS
(T_g = 2465 F abs, T_c = 1000 F abs.)

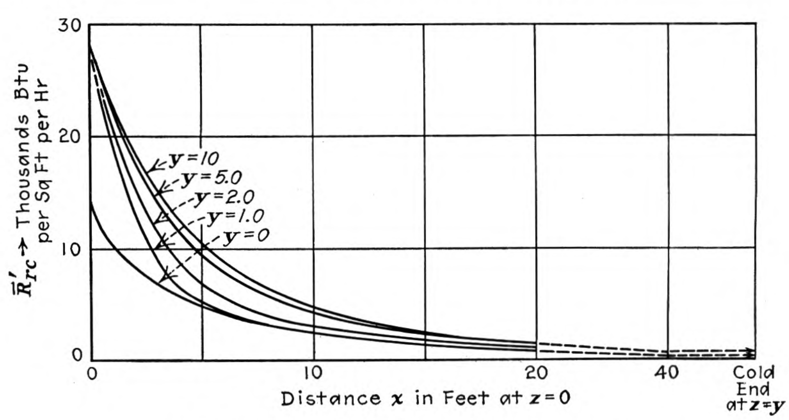


FIG. 5 $\bar{R}'_{rc}$, REFRACTORY WALL ⇆ COLD WALLS
(T_r = 2540 F abs, T_c = 1000 F abs.)

plain. Curve $y = 1.0$ does not peak because the combination of the rapid drop in the refractory curve for $y = 1.0$, Fig. 5, plus the effect of decreasing surface of burning particles more than offsets the sum total of other influences which tend to increase the absorption rate as the surface point moves away from the head wall. For $y = 0$, however, the curve again peaks because, as shown in Fig. 5, the contour for $y = 0$ starts low and so its gradient is much smaller than is that for $y = 1.0$. It follows that, for the conditions of this problem, the total contours between $y = 0$ and $y = 1.0$ except possibly those for which y is a very small quantity Δy, fall from a high point at $x = 0$ to a low point at $x = 40$ without peaking.

Relations for Determination of Radiant-Energy Absorption at Points on Cold Walls of Cavity. At any point of cold surface the radiant-energy absorption, Btu per sq ft per hr is

$$\overline{R}'_c = \overline{R}'_{uc} + \overline{R}'_{jc} + \overline{R}'_{gc} + \overline{R}'_{rc} \qquad [1]$$

where index prime denotes this basis of reference. Then, with the same notation[1] except where otherwise noted

$$\left.\begin{aligned} \overline{R}'_{uc} &= \sigma\, \bar{\mu}'_{uc}\, F'(T_u^4 - T_c^4) \\ \overline{R}'_{jc} &= \sigma\, \bar{\mu}'_{jc}\, F'(T_j^4 - T_c^4) \\ \overline{R}'_{gc} &= \sigma\, \bar{\mu}'_{gc}\, F'\,(T_g^4 - T_c^4) \\ \overline{R}'_{rc} &= \sigma\, \bar{\eta}'_{cr}\, F_{cr}(T_r^4 - T_c^4) \end{aligned}\right\} \qquad [2]$$

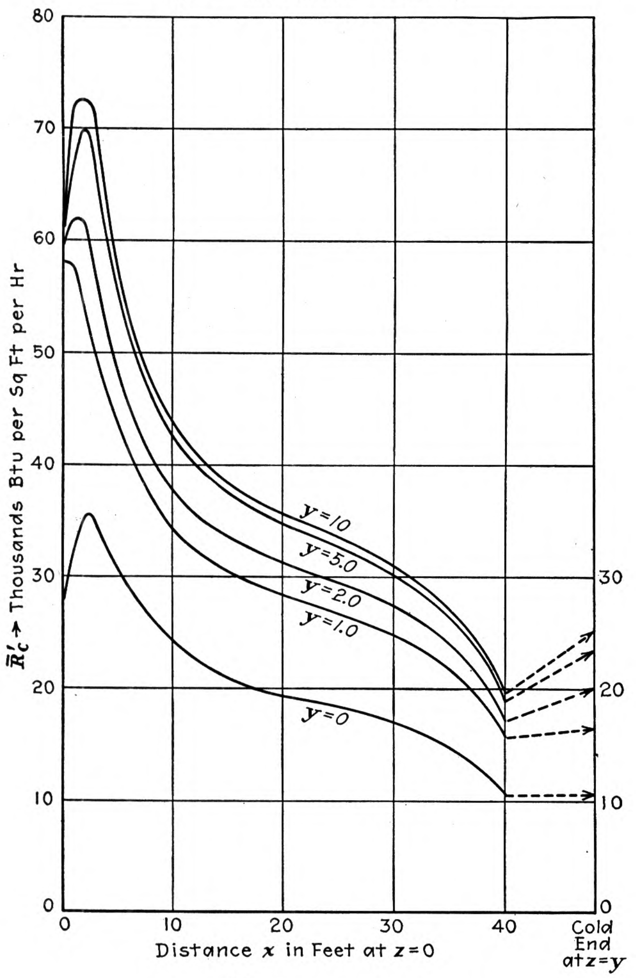


FIG. 6 $\overline{R}'_c$, TOTAL OF FIGS. 2 TO 5

FIGS. 2 TO 6 CONTOURS OF ENERGY ABSORPTION AS HEAT OVER COLD WALLS OF CAVITY
(Coordinates x, y, and z and furnace conditions as in Fig. 1.)

Angle factor F' is unity at all wall points except at an edge where two wall surfaces meet, or in a corner where three wall surfaces meet. For rectangular parallelepipeds $F' = 0.50$ along edges and 0.25 in corners. The special form on the right for $\overline{R}'_{rc}$ follows from the relation

$$A'_c F'_{cr} = A_r F'_{rc} \qquad [3]$$

where cold surface A'_c is now unity. It is noted also that $\bar{\eta}'_{cr} = \bar{\eta}'_{rc}$.

Emissivity coefficients $\bar{\mu}'$ and transmissivity coefficient $\bar{\eta}'$ are each radiant means with respect to the point of cold surface under consideration. As an example, let μ' represent the value of the emissivity coefficient for differential region subtending angle factor dF from the considered point of cold surface. Then

$$\bar{\mu}' = \frac{1}{F}\int_0^F \mu' dF \qquad [4]$$

where F is the angle factor of the total region from which the radiation under consideration may be thought of as converging toward the point under consideration. For an element dF of this region

$$\left.\begin{aligned} \mu'_{uc} &= (1 - \psi'_j f'_{jc})\,(1 - G''_{uc}) f'_{uc} \\ \mu'_{jc} &= (1 - \psi'_u f'_{uc})\,(1 - G''_{jc})\, f'_{jc} \\ \mu'_{gc} &= G'_{gc} - f'_{gc}\,(G'_{gc} - G''_{pgc}) \\ \eta'_{cr} &= (1 - f'_{cr})\,(1 - G'_{cr}) \end{aligned}\right\} \qquad [5]$$

Here ψ'_j represents the probability that ash particles j will fall between burning particles and cold-surface point c. Factor ψ'_u is the same thing for burning particles u, relative to ash particles and cold surface. For uniform distribution relative to each other for the particles u and j, $\psi'_u = \psi'_j = 0.50$. If particles u and j are completely confined to separate zones with respect to each other along the radiant beam through both sets of particles, and radiation is from, say, particles u through particles j to wall point c, then $\psi_j = 1.00$. If under the same conditions radiation is from particles j through particles u to the wall, then $\psi_u = 1.00$. In view of these statements it is obvious that the following values of ψ apply to the conditions in the present furnace cavity:

All transverse beams, $\psi_u \approx \psi_j \approx 0.5$.

Radiation u to refractory end, $\psi_j \approx 0$.

Radiation j to refractory end, $\psi_u \approx 1.00$.

Radiation u to cold surfaces, ψ_j is between 0.5 and 1.00 and the mean value for radiation to all cold points is closer to 1.00 than to 0.5.

Radiation j to cold surfaces, ψ_u is between 0 and 0.5 and the mean value for radiation to all points is closer to 0 than to 0.5.

Factors G'' in Equations [5] are similar to factors G' but the beam length l'' varies in value between 0.5 l' and l' depending on the relative distribution between the intervening class of particles u, j, or p and the radiating gas molecules. The following range of values is seen to apply for the furnace conditions in the present cavity:

For G''_{uc} and point c near head end of cavity, $l'' \approx 0.5\, l'$. For point c at cold end $l'' \approx l'$.

For G''_{jc} and any location of point c, $l'' \approx 0.5\, l'$.

For G'_{pgc} all particles are considered as in one class p, and $l'' \approx 0.5\, l'$.

In other respects the notation of Equations [5] is similar to that employed in the paper.

In order to see how information, on which values of ψ', f', G', and G'' depend, may be furnished in such form that the results for $\bar{\mu}'$ and $\bar{\eta}'$ approximate those for radiant means of the type illustrated by Equation [4], consider the geometry shown in

Fig. 7, which applies to rectangular parallelepipeds. Here p is the surface point under consideration. For a given set of furnace conditions the radiant-mean values of ψ', f', G', and G'' throughout plane section *pqrs*, with respect to point p, may be approximated by reference to: (1) The radiant-mean beam length from p throughout the plane section; and (2) the line-mean values along x from x at p to x at q, of the concerned particle areas a_u, a_j, or a_p and of the concentration v of radiating gas molecules.

Of these line-mean particle areas between x at p and x at q, Fig. 7, that for a_u applies to f'_{uc}, that for a_j applies to f'_{jc}, and

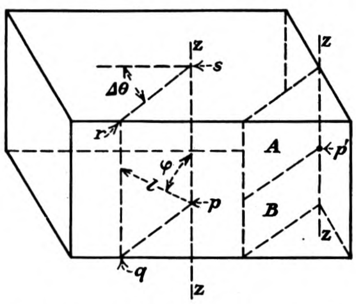


FIG. 7 GEOMETRICAL AIDS IN APPROXIMATING RADIANT-MEAN VALUES WITH RESPECT TO A POINT ON A WALL

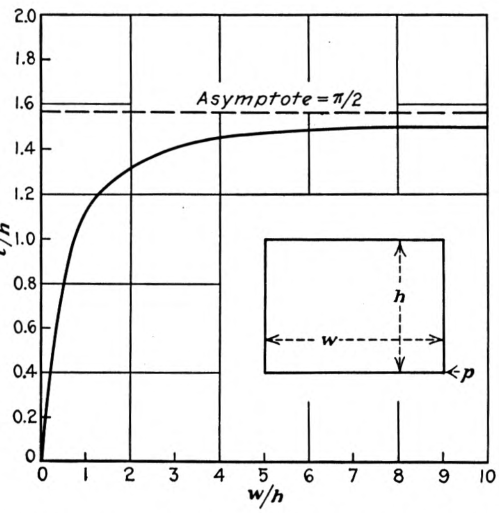


FIG. 8 MEAN BEAM LENGTH l' WITH RESPECT TO A CORNER p FOR A RECTANGULAR PLANE SECTION

that for a_p to both f'_{gc} and f'_{cr}. The mean beam length l' from a corner p of plane section *pqrs* is, of course

$$l' = \frac{2}{\pi}\int_0^{\pi/2} l \cos\varphi \, d\varphi \qquad [6]$$

which on integration may, for convenience in use, be represented as shown in Fig. 8.

If p is located in the cold end of the cavity as at p', Fig. 7, the plane section is divided into parts A and B. The mean beam length l' for the whole section is a simple average of those for the two parts. Thus it is seen how values of ψ', f', G', and G'', and hence those of μ' and η' may be approximated for plane section *pqrs* in any given position of rotation θ.

Suppose for a moment now, that instead of approximating μ' as just outlined, it had been evaluated on the basis of the relation

$$\mu' = \frac{2}{\pi}\int_0^{\pi/2} \mu \, d\varphi \qquad [7]$$

where the emissivity coefficient has value μ along any direction l in section *pqrs*. This is its exact radiant-mean value for this plane section. On this basis the value of $\bar{\mu}'$ as found by means of

$$\bar{\mu}' = \frac{1}{\theta}\int_0^{\theta} \mu' \, d\theta \qquad [8]$$

where edge qr of the plane section slides along the walls of the cavity, is exactly equal to that found on the basis of Equation [4]. Hence, if μ' in Equation [8] is determined on the approximate basis before outlined instead of by means of Equation [7] then Equation [8] yields a value of $\bar{\mu}'$ which closely approximates that found on the basis of Equation [4].

The integration indicated in Equation [8] is of course seldom possible or feasible for actual furnace conditions. In lieu of this, let plane section *pqrs* occupy successive positions θ spaced by equal intervals $\Delta\theta$ from each other and covering the region θ from which radiation converges to the point p. On this basis the value of $\bar{\mu}'$ is a simple average of values found for each of the successive positions of *pqrs*.

It is noted that this procedure may be employed to find the radiant-mean values $\overline{\psi'f'}$, $\bar{G}'$, and $\bar{G}''$ of the coefficients $\psi'f'$, G', and G'' which appear in Equations [5]. When this is done the values $\bar{\mu}'$ and $\bar{\eta}'$ are computed from Equations [5] with $\overline{\psi'f'}$, $\bar{G}'$, and $\bar{G}''$ substituted for $\psi'f'$, G', and G''. They are thus products of the radiant means of the involved factors instead of the radiant mean of the product of such factors. This of course superimposes an additional approximation. The error involved is, however, a small one in most cases and the procedure is less cumbersome when this is done. For these reasons it was employed.

Additional Omissions and Corrections. In Equation [34], angle-factor term F_{ur} was omitted from both numerator and denominator. The corrected equation is thus

$$v_{ur} = \frac{\dfrac{1}{\varphi_z}\displaystyle\int_0^{\varphi_z} v_\varphi a_u F_{ur} \, d\varphi}{\dfrac{1}{\varphi_z}\displaystyle\int_0^{\varphi_z} a_u F_{ur} \, d\varphi} \equiv \frac{\overline{v_\varphi a_u F_{ur}}}{\overline{a_u F_{ur}}} \qquad \text{for [34]}$$

In Equation [43] the radiation coefficient σ should appear in the denominator. Thus

$$G_{12} = \frac{\pi - \pi_2}{\sigma(T_1^4 - T_2^4)} \qquad \text{for [43]}$$

Beam-length values for $\bar{\bar{l}}_{ur}$, $\bar{\bar{l}}_{uc}$, $\bar{\bar{l}}_{ug}$, $\bar{\bar{l}}_{jr}$, $\bar{\bar{l}}_{jc}$, and $\bar{\bar{l}}_{jg}$ in Equation [40] should have been computed on the basis that the source of the radiation is composed of *spherical* particles scattered uniformly throughout the radiant-mean plane for the class of particles indicated by the subscript. The beam length is then between this source just defined and the boundary indicated by the second subscript.

On this basis the following beam-length values should replace corresponding values in Equations [40] and [42].

$$\bar{\bar{l}}_{ur} = 3.7 \text{ ft} \qquad \bar{\bar{l}}_{uc} = 13 \text{ ft} \qquad \bar{\bar{l}}_{ug} = 9.0 \text{ ft}$$

$$\bar{\bar{l}}_{jr} = 14.5 \text{ ft} \qquad \bar{\bar{l}}_{jc} = 13.5 \text{ ft} \qquad \bar{\bar{l}}_{jg} = 13.7 \text{ ft}$$

$$\bar{\bar{l}}_{upg} = 4.5 \text{ ft} \qquad \bar{\bar{l}}_{jpg} = 6.8 \text{ ft}$$

On the basis of Equation [34] as revised and the corrected beam lengths, the corrected data for the equilibrium solution are as before indicated in this discussion.

Equation [57*b*] should be

$$\bar{R}_{rj} + \bar{R}_{uj} + \bar{R}_{gj} + \bar{L}_{gj} = 17 \text{ per cent} \qquad [57b]$$

Term $\bar{R}_{rj}$ was omitted.

Combination Oil-and-Gas Burners[1]

H. H. Wellander.[2] The writer would like to have the author explain the method he uses to correct for sulphur dioxide in gas samples. In the burning of refinery gases rich in hydrogen sulphide, a considerable amount of sulphur dioxide is found in the flue gases. In an ordinary gas analysis, the carbon dioxide and sulphur dioxide are both measured as carbon dioxide. Therefore, the author should explain whether or not his curves are based on a sulphur-free determination of the flue gases.

D. S. Frank.[3] The author's paper presents a timely comprehensive picture to the engineer interested in fuel. With the advent of low-priced natural gas, which is available in various territories, and the extra incentive of having burner equipment capable of taking natural gas at periods of low demands on the gas-company's distribution system, the combustion engineer is faced with the problem of selecting proper fuel stand-by equipment. This stand-by equipment must be for either pulverized coal or fuel oil, depending on the cost per 1,000,000 Btu.

The paper points out very clearly the main factors which should be taken into consideration when selecting the combination oil-and-gas burners.

The prime function of any oil burner is to properly atomize the fuel oil. Whether the atomization is obtained by pressure, as in the mechanical burner, or by steam or air, the important point is to break the oil up into particles as minute as possible. The next function is to surround properly each minute globule of oil with sufficient air to burn the oil completely, quickly, and with a minimum amount of excess air.

The problem of controlled flame length and efficient combustion is primarily one of turbulence. The importance of this one factor assuming sufficient time and temperature are present, cannot be overestimated.

Generally speaking, the greater the turbulence the shorter the flame, but also a noticeable increase in flame width on expansion through the burner throat. In view of this, the design of the combustion chamber tends to limit the choice of burner. In a long narrow firebox, the burners shown in Figs. 8, 9, and 10 of the paper would be more conducive to lower wall maintenance than the burners shown in succeeding figures. However, it appears that the burner illustrated in Fig. 16, due to the adjustable feature of the gas jets and individual air control, lends itself to practically any combustion-chamber design. This adjustable feature with ability to obtain various flame contours should tend to minimize burner troubles which have heretofore existed.

The proper selection of burner tips is also important. The writer has found that the multiple-round-hole tip is more conducive to low excess air when used with an air register than the flat flame tip. In a burner employing multiple gas jets, either the multiple-round-hole tip or the flat tip can be used with equal success. Here again, the design of the combustion chamber and spacing of the required number of burners tend to fix the correct type of tip to be used.

Ordinarily, the greater the turbulence, the higher the draft drop across the burner. In a forced-draft installation, this point can be taken care of in the fan and duct design, but in designing natural-draft installations it must not be overlooked.

Draft loss through the vanes and throat of an air-register type burner is a function of Btu release—the higher the Btu release, the higher the draft loss. For example, firing natural gas with 20 per cent excess air, the draft loss at a release of 10,000,000 Btu per hr is 0.19 in. of water, whereas at a release of 14,000,000 Btu per hr, the draft loss increases to 0.41 in. of water. Draft loss is also a function of throat diameter and the values just given by the writer are for a definite throat size. At the same Btu release on a larger throat diameter, the draft loss would be less than the values given. However, with increase in throat diameter, there is a decrease in turbulence for the same Btu release.

In burning fuel oil, the draft drop changes with the atomizing medium, as the draft loss is less for a mechanical-atomizing burner than for steam-atomizing burner, comparison being made at the same Btu release.

When designing the wind box for a forced-draft installation, considerable thought should be given to the capacity on natural-draft operation. If quickly detachable plates are incorporated in the wind-box design, and friction drop is low, the forced-draft burner can be operated on natural draft within reasonable limits. Longer flame lengths will, of course, be obtained.

The writer is familiar with the author's combustion charts and has found them very useful in making rapid furnace-efficiency calculations.

It is interesting to note that the stack-loss chart for a 10 A.P.I. residual petroleum oil can be used with very little error on any gravity fuel oil from 0 A.P.I. to 40 A.P.I. Assuming an arbitrary condition of 600 F stack temperature and 10 per cent excess air, a 0 A.P.I fuel oil shows a stack loss of 17.5 per cent; a 10 A.P.I. oil shows a stack loss of 17.7 per cent; and a 40 A.P.I. oil shows a stack loss of 18.5 per cent. Therefore, under the foregoing conditions, a maximum error would exist of 1.0 per cent. At 100 per cent excess air, the stack loss varies from 26.5 per cent for a 0 A.P.I. oil to 27.3 per cent for a 40 A.P.I. oil, or a maximum error of 0.8 per cent is involved.

Since industrial fuel oil is within the previously mentioned gravity range, and normally the excess air is also within this range, the chart can be used for rapid furnace calculations with a negligible error.

The stack-loss charts for a combination of fuels have proved to be helpful in determining furnace efficiency, and they are recommended for plants firing more than one fuel simultaneously.

R. C. Vroom.[4] In connection with the charts the author has prepared, perhaps it might be well to emphasize that those giving heat losses are in all cases based on complete fuel combustion.

Small furnaces and high rates of heat release, as the author points out, require efficient burners. The writer has fired with oil at a rate of 1300 lb of bunker fuel per burner per hr with excellent efficiency into a boiler furnace where the heat release approached 1,000,000 Btu per cu ft hr. In the U. S. Navy, heat releases exceeding 300,000 Btu per cu ft per hr with oil fuel are standard practice and the boiler efficiency is excellent. Our European neighbors are ahead of us in putting into practice in stationary plants what is possible in the way of furnace-heat releases. There is no valid reason why similar heat releases are not feasible with gas fuel using combined burners.

The author has in a sense defined the combined gas-and-oil burner as one having the ability to burn oil or gas equally well, separately or in combination. The words "equally well" are vital in this definition because there are few burners which fulfill that part of it. There are many which burn gas with a reasonable degree of economy; however, the oil feature is suitable only for stand-by purposes. In certain of these, when shifting from gas to oil, the output must be reduced and the excess air greatly increased. With some designs the use of oil alone for extended periods results in irreparable damage to the gas-burning elements.

[1] Published as paper FSP-60-14, by O. F. Campbell, in the August, 1938, issue of the A.S.M.E. Transactions, pp. 457–467.

[2] Chief Power Engineer, Los Angeles Works, The Texas Company.

[3] Assistant Chief Combustion Engineer, The Pure Oil Company, Chicago, Ill.

[4] Chief Engineer, Peabody Engineering Corporation, New York, N. Y. Mem. A.S.M.E.

The mere ability to burn oil temporarily by no means qualifies such burners as combined burners.

The ability of a burner to operate without cleaning is of major importance where oil-refinery heaters are fired with wet gas, especially if no distillate trap is installed. Several of the special designs for refineries shown in the paper, including the open-ring burner shown in Fig. 14 (in which the operating mechanism now has been modified to avoid the mechanical difficulty mentioned), are in general applicable to one particular kind of refinery heater.

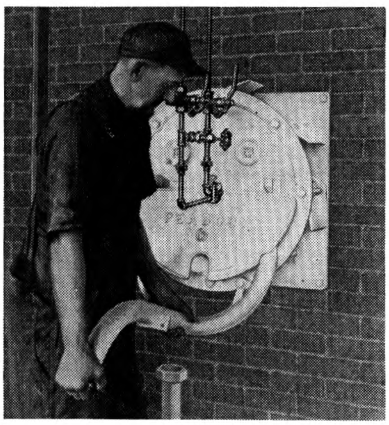


FIG. 1

However, the open gas ring which has no restrictions to cause plugging can be used with a standard air register and this assembly has been employed for firing boilers and many types of oil heaters. The removable closed-ring burner design, shown in Fig. 1 of this discussion, has found wide application in many refineries because it permits replacement of a plugged gas ring and at the same time permits oil to be burned during replacement.

In regard to liquid fuels, it is stated in the paper that any grade of fuel oil which can be delivered to a burner can be burned satisfactorily if the burner design is correct. This is true, but it is important in selecting a burner that full consideration be given to all the fuels which will have to be burned and to the results which will be considered as constituting satisfactory operation. A distillate burner, for example, could not be considered of incorrect design because it will not handle bunker *C* fuel. Neither should a burner be termed unsatisfactory if a fuel contains salts which attack the furnace brickwork. Sometimes such difficulties as unforeseen steam temperature, tube failure, high excess air in the exit gases, high stack temperature, low efficiency, and pulsation are due to faulty burner design or operation. Not infrequently the amount of superheater surface, scale in the tubes, leaky settings, fouled heating surface, and insufficient draft are largely responsible for such troubles.

In boiler firing, a combined burner is gaged by its ability to operate indefinitely with either fuel without damaging the elements provided for burning the other, and to do this over the full capacity range with complete combustion and a minimum of excess air. At the same time, the burner must be of a type which permits changing from one fuel to another almost instantaneously. Fig. 2 of this discussion shows a type of combined burner which is giving an excellent account of itself in most of the modern gas-and-oil-fired utility plants, as well as in many industrial plants and refineries, on the Pacific Coast. There are 225 burners in one station in Los Angeles. These plants are firing natural and refinery gases, bunker oil and refinery sludges, including acid sludge. For liquid fuel, some plants employ steam atomization while others use mechanical atomization. All types of boilers are being fired including integral-furnace and other recent designs.

In referring to the burner shown in Fig. 12 of the paper, the author mentions its lack of flexibility when burning oil due to mechanical fuel-oil atomization. Many engineers are of the opinion that all mechanical atomizers are greatly limited in capacity range, because such is the case with most of them. However, Estcourt[5] and others, who have had operating experience with the type of burner he describes, which is that shown in Fig. 5 of Philo's paper,[6] realize that a mechanical atomizer can have an extraordinary degree of flexibility.

In Estcourt's paper,[5] which describes what has been adopted as the regular low-load operating procedure at Station *A* of the Pacific Gas and Electric Company, San Francisco, Calif., it is shown that the automatic-control equipment is required to operate satisfactorily over a load range between maximum and less than 5 per cent of maximum capacity. With the 1400-lb boilers, each having a steam-generating capacity of 500,000 lb per hr, the

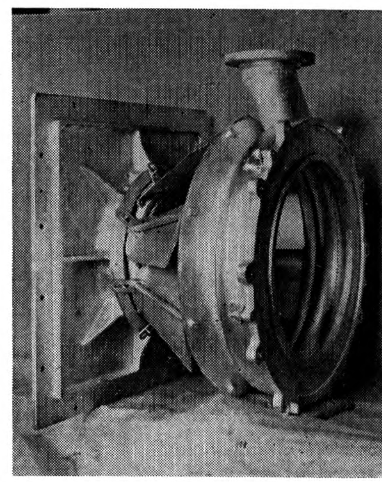


FIG. 2

minimum steam flow for satisfactory continuous operation with all burners in service is approximately 30,000 lb per hr (17-to-1 range) for gas fuel, and 65,000 lb per hr (8-to-1 range) for oil fuel. This paper[5] also states that it is entirely possible to operate (on swings) down to the same minimum load (oil fuel) with all burners in service (17-to-1 range) as is obtainable with gas fuel. In other words, the mechanical atomizers in this plant are anything but inflexible.

The paper[5] also states that "it is entirely possible that still lower loads may be obtainable. . . .(but) the figures just given are based upon a conservative opinion as to what is a satisfactory operating condition in the furnace." The burner gas pressure was 8 lb per sq in. at maximum capacity, and anyone who is interested can readily calculate what pressure would have been required for the minimum capacity. The arrangement for firing oil employed is essentially the constant-differential method of control mentioned by Philo in his paper.[6]

AUTHOR'S CLOSURE

In reply to Mr. Wellander's question regarding the flue-gas analysis to correct for sulphur dioxide in flue-gas samples, it will be necessary to know the refinery gas analysis and calculate the SO_2 produced as CO_2 and the stack-loss chart calculated on this basis. The CO_2 curve will be the CO_2 plus SO_2.

[5] "Design and Operating Problems With Gas- and Oil-Fired Boilers for Stand-By Steam-Electric Stations," by V. F. Estcourt, Trans. A.S.M.E., vol. 59, January, 1937, paper FSP-59-1, p. 7.

[6] "Technique of Burning Fuel Oil and Natural Gas," by F. G. Philo, *Mechanical Engineering*, vol. 60, April, 1938, p. 315.

The Behavior of Sodium Sulphite in High-Pressure Steam Boilers[1]

H. F. Johnstone.[2] It is important to note that there is no essential disagreement between the results reported by the authors and those of Taff, Johnstone, and Straub.[3] The tests made by the latter were run at concentrations of about 4000 ppm of sodium sulphite. Under these conditions decomposition was found in every case at temperatures of 545 F and higher. These results have been verified by later tests made by Professor Straub using bombs under steaming conditions. There were two reasons for choosing the high initial concentration for the tests: (1) To explain the disappearance of sodium sulphite from a high-pressure boiler in which localized concentration was believed to exist; and (2) to permit following the reaction without using complicated semimicro analytical methods.

At the high concentrations the rate of decomposition is proportional to the concentration of residual sulphite, a relationship which is generally said to indicate a first-order reaction. In all of the tests made over a 36-hr period, however, the percentage of sulphite remaining was between 2.4 and 4.1 per cent of the original amount, corresponding to 96 to 160 ppm. Since this concentration would have been reached in a much shorter time (6 to 23 hr at 600 to 550 F), if the initial rate of decomposition had prevailed, it is evident that the reaction either reached equilibrium, or slowed down appreciably at low concentrations. The small concentrations of sulphide found in high-pressure boilers by Hitchens and Purssell,[1] using sensitive methods of analysis, show that the latter alternative is probably true. The results of the two investigations,[1,3] therefore, are not at variance and, since they were carried out under different conditions, may be considered as actually confirmatory.

The fact that the decomposition is "pseudounimolecular" rather than of true first order is only of academic interest. The practical engineer is primarily interested in knowing the answer to the question: "Does sodium sulphite decompose in actual operation?" On the evidence obtained so far, the answer is: "Yes, at appreciable rates at temperatures above 545 F and concentrations above 200 ppm. At lower concentrations, not exceeding 30 ppm and in a boiler in which no abnormal concentrations are likely to occur, the salt does not decompose at a rate to be of practical importance."

Even when decomposition does occur, little is known about the nature of the products and whether their presence would be considered detrimental to boiler operation. It would be expected that the hydrolysis of sulphides to give hydrogen sulphide in the steam would take place as readily as that of carbonates to give carbon dioxide. There is indeed some evidence that this hydrolysis does take place. First, in the laboratory tests under steaming conditions the total sulphur content decreases when the vapors come in contact with copper or clean iron surfaces. Second, in four of the tests reported by the authors[1] on high-pressure boilers, no sulphide was found in the boiler water but corresponding analyses by their sensitive methods showed sulphide in the saturated steam entering the superheater, before concentration of carry-over could have taken place.

It is obvious that the chemical investigation of this subject is not complete. In view of the complexity of the reactions of sulphur compounds as shown by the extensive work of Foerster on decomposition of sulphurous acid and its salts, further investigation would seem to be justified as a precautionary measure.

L. Drew Betz.[4] The use of sodium sulphite for oxygen removal has assumed a prominent place of importance in the field of water-conditioning chemistry. The behavior of this salt at elevated temperatures and under pressure is therefore of considerable importance. Prior to the papers presented by Taff, Johnstone, and Straub[3] and the authors,[1] little was known of the action of sodium sulphite under boiler-operating conditions.

The authors in the paper under discussion present some very interesting data and are to be congratulated upon the thoroughness of their investigation. Particularly impressive are the methods of analysis they have worked out for the determination of sulphides and sulphites in minute quantities in condensed steam. Without having actually used the methods proposed by the authors, it is very difficult to present a constructive discussion. With such information available, however, it should lend impetus to additional research in this field.

Regarding test method *A* which the authors describe in this paper, no mention is made of the possible interference of iron. Snell,[5] in describing a method he proposes for sulphides based on the same reactions given by the authors, states that while small amounts of iron do not interfere, larger quantities may possibly cause some trouble. Also, in using this method for the determination of sulphides in boiler water, some difficulties might ensue if organic matter be present in sufficient quantities to cause a discoloration of the water. Such difficulties would be in the nature of matching the color produced with the color standards.

It would also be of material value to ascertain whether any reactions occur between sodium sulphite and organic matter at elevated pressures. An analysis, made some few months ago in the writer's laboratories, of a deposit from a boiler tube, showed the presence of sulphates in quantities greater than the amounts necessary to combine with the inorganic cations present. The deposit was high in organic matter, leading to the assumption that the sulphate was originally present in the form of sulphite and quite possibly combined with the high organic content. A reduction in the amount of sodium sulphite used, in this instance, eliminated further deposits of this character.

Authors' Closure

The authors agree with Professor Johnstone that there need be no essential disagreement between the results reported by himself and coworkers and those by Hitchens and Purssell since the tests were made under entirely different conditions.

It is still felt that the traces of sulphite and sulphide found in the steam of the two higher-pressure boilers mentioned in the paper did not originate from hydrolysis of sulphites and sulphides in the boiler water but rather from decomposition of sodium sulphite carried over mechanically into the steam. The evidence for this is that the sulphite and sulphide both disappeared completely in the steam with decreased load on the boilers, a situation which is untenable with hydrolysis which proceeds at a rate quite independent of the steaming rate.

The authors still feel that the extent of contamination of the steam is too minute to be of any practical significance.

It is true as Mr. Betz says that appreciable amounts of ferric iron interfere with the colorimetric determination of sulphide with lead plumbite in method *A*. In fact the two react as follows

$$2\,Fe^{+++} + S^{=} \rightarrow 2\,Fe^{++} + S$$

[1] Published as paper FSP-60-15, by R. M. Hitchens and J. W. Purssell, Jr., in the August, 1938, issue of the A.S.M.E. Transactions, pp. 469–473.

[2] Associate Professor of Chemical Engineering, University of Illinois, Urbana, Ill.

[3] "Decomposition of Sodium-Sulphite Solutions at Elevated Temperatures," by W. O. Taff, H. F. Johnstone, and F. G. Straub, Trans. A.S.M.E., vol. 60, April, 1938, pp. 261–265.

[4] General Manager, W. H. & L. D. Betz, Philadelphia, Pa. Mem. A.S.M.E.

[5] "Colorimetric Analysis," by F. Dee Snell, D. Van Nostrand Company, Inc., New York, N. Y., 1921.

There was too little iron in the samples of condensed steam to be of importance.

In the boiler water the presence of ferric iron is improbable since the excess sodium sulphite present would reduce it to the ferrous condition. It is felt that iron did not interfere with the tests for sulphide in the boiler water since addition of definite amounts of sulphide gave colors comparable to those with distilled water, iron-free.

It is true of course that the method is applicable only to a clear boiler water. Those the authors encountered were clear. It should be feasible to filter the water before applying the test with no loss of sulphide since the excess sulphite would keep it in the reduced form. Discoloration caused by organic matter might be circumvented by removal of the sulphide with nitrogen in a faintly acid solution and collecting on a lead-acetate strip. Each such water would be a research problem in itself.

The authors have no information on the reaction of sulphite with organic matter in boiler water at high temperatures and pressures. It will react quantitatively with aldehydes and ketones at lower temperatures. The authors are much interested in Mr. Betz's findings and would appreciate hearing of any later developments in the problem.

An Oil-Bath-Lubricated Railway Bearing[1]

H. M. Warden.[2] The writer has studied prints and has seen the "Disc-Flo" journal units in operation. There are several features that are improvements over the conventional box and parts, some of which would greatly improve the performance of present bearings if they were incorporated in A.A.R. designs.

One such item is dust guards. The old wood dust guard universally used insures no certainty of oil supply as it will not prevent the loss of oil. It is frequently detrimental to good bearing and journal performance as it fails to keep water and dirt out of the box. They make a poor fit between the box and the axle, and are not moisture-proof.

The paper in my opinion is correct in its statement that "serious trouble experienced today with A.A.R. type of bearing and box is not friction." Journal friction when reduced to a minimum by use of suitable oil and waste, properly turned journals, and broached bearings is but a small part of the total rolling friction. If oil, waste, journals, or bearings are not correct, hotboxes are apt to occur; however, failures of these parts are frequently due to presence of foreign matter in the box. A poor grade of oil or unstable feeding of oil, as well as presence of dirt in the box, are causes of most failures.

A good dust guard should provide an oil seal around the axle and in the dust-guard cavity, and the design should readily fit present A.A.R. boxes. It should also be manufactured at a reasonable cost.

Another item is the necessity of providing for lateral thrust of the axle to be applied against the fillet of the journal bearing by the journal fillet. A majority of bearings show excessive wear on the collar, which, besides loosening the lining, increases lateral play on a truck which already has too much. With present A.A.R. trucks operating under bad conditions, such as poor track, it is possible for the back of the boxes to strike the hub of the wheel. Changes necessary to locate lateral thrust at the journal fillet can be made by reducing tolerances and changing dimensions of bearings and wedges.

The self-aligning fit between the wedge and the roof of the box will overcome any irregularities in the box or contained parts, but the writer believes if it were exaggerated as shown in this paper it would further cut down lateral play since it would take the place of the wedge stop. Lack of oscillating movement for which the wedge is intended will cause the brass to bind, producing uneven loading, excessive friction, and eventually a hot bearing. When the top bearing is untrue it gives uneven pressure on the journal on account of improper seating.

The writer does not believe that a flat-back bearing or that extending the sides of the bearing will materially eliminate bearing tilt caused by impacts, but would probably take care of side movement resulting from reasonable brake applications if the boss in the box were extended close to the center line. Heavy impacts and emergency air-brake applications, even at slow speed, cause the bearing, wedge, and box to raise momentarily off the journal. If impact be heavy enough to displace the packing, which may be caught under the bearing, the result will be waste grabs.

The use of hardened pedestal liners is desirable to minimize wear. When pedestals are worn, a jerky movement occurs between the box and the pedestal and rocking of the box produces a distorted bearing face in the lining. Excessive wear may cause partial locking of the box. A pedestal out-of-alignment may prevent proper seating of the bearing and produce collar friction.

If recognized methods of maintenance and application of parts be closely followed and suitable oil and waste used, the A.A.R. journal box and parts will give satisfactory service; however, if the A.A.R. would adopt some changes in design incorporated in the Disc-Flo unit there is no question but that we would obtain longer life at less maintenance and have less trouble with journal box and contained parts.

In the foregoing comments the writer has in mind freight equipment, whereas the use of the Disc-Flo journal unit as covered by this paper seems to be intended for passenger-car service and would necessitate carrying a different stock of parts at all points for protection to such equipment. It is true that our passenger-car boxes and parts, wheels, and axles are better maintained and inspected more frequently, yet at times they operate under severe conditions. The writer understands that very few units have been applied for service tests and is not familiar with such tests, but he believes additional long-time service tests should be made under all conditions so that reasonably accurate comparative figures with A.A.R. standards would be available.

G. W. Ditmore.[3] The writer is not in a position to discuss the Disc-Flo unit from a practical standpoint having had no experience with its operation other than laboratory demonstrations. Laboratory tests have shown this device to possess great possibilities toward improved car-journal lubrication, providing actual service conditions do not develop unlooked-for difficulties.

The problem of developing a satisfactory sealing arrangement at the back of the box, that will function properly for the service life of the wheels without constant policing, when solved should be of great economic value to the railroads.

The authors deserve commendation for the exacting and thorough manner with which they have approached and are working to overcome one of the difficult problems of railroad operation.

The operation of Disc-Flo lubrication units in actual service will be followed with keen interest.

Frank E. Cheshire.[4] The writer has observed much of the laboratory research and many of the service tests of the Disc-Flo journal-bearing assembly. These observations have been made

[1] Published as paper RR-60-3, by Albert Vigne and I. E. Cox, in the August, 1938, issue of the A.S.M.E. Transactions, pp. 499–506.

[2] Chief Mechanical Officer, Missouri-Kansas-Texas Lines, Parsons, Kan.

[3] Master Car Builder, Delaware & Hudson Railroad Corporation, Albany, N. Y.

[4] General Car Inspector, Missouri Pacific Lines, St. Louis, Mo.

as a railroad mechanical man. It is apparent, from approximately two years of operation under varying but representative service conditions, that a constant supply of lubricant is delivered to the contact surfaces by the device described by the authors. The problems encountered throughout these tests have not involved the lubrication as such. Alignment, control of the axle lateral play sealing the oil in and extraneous contamination out, and the difficulties arising therefrom have required correction. As long as oil has been available—and with one exception that condition has obtained—the requisite lubrication has been provided. That one exception developed the prime necessity for closer tolerances on axles new and reworked, on bearings, boxes, wedges, and truck dimensions. In this one failure the oil supply was lost due to a combination of wide tolerances permitting excessive uncontrolled axle lateral play. This condition is common in the conventional assembly and accounts for much of the loss of oil and entrance of extraneous contamination, in addition to excessive wear of the various parts of the assembly and resultant limited service life.

Since incorporation of corrective tolerances, several thousands of miles of satisfactory service indicates that this assembly will provide dependable service. This indication will be proved or qualified as the service mileage increases.

The writer believes that the inclusion of the research data in presenting this device offers a valuable contribution to advance in the science and art of journal lubrication. Speaking as a railroad mechanical man, it is encouraging to see presented a logical analysis of the fundamentals of one of our major problems. In something more than 20 years of almost daily contact with failures of journal lubrication, many of the phenomena, here broken up into causal components, have been observed. The writer has observed in service an incorporation of many of the corrections offered. The encouraging results therefrom induce a rather critical conviction that consistent improvement of considerable economic worth may justifiably be expected from an incorporation of the recommended corrections in the fundamentals of design and standards of practice. It is the writer's observation that our most difficult problems arise from the necessity for correcting faults in existing materials. The inexorable characteristics of established standards and practices have the weight of years of use. No one factor of railroad mechanical operation is more formidably surrounded by tradition than that of journal lubrication. Each railroad has followed a practice dictated by its individually traditional standard. It appears that efficiency in journal lubrication is measured by comparative reductions in transportation delays; or, in the words of an able contributor to critical investigation, "miles per hotbox is the operating bible." Such a yardstick does not include the greater economic factor, that is, the cost to attain that particular record.

It is the writer's further opinion that critical study and analysis of the data presented by the authors, and correlation thereof with service indications evident to anyone interested, will do much to remove prejudice, eliminate weak imitation, and discard outworn tradition in relation to journal lubrication. It occurs to the writer that the formulas presented by the authors for the determination of actual axle lateral movement, under any given set of dimensions, is a constructive contribution. The necessity for a definite imposition of the thrust load at the fillet end of the bearing should be obvious but has not been generally recognized. The means suggested for control of the axle lateral movement, within reduced limits, is simple and effective. Its effectiveness has been proved in service.

In the formulas, the truck dimension indicated as the inside flanges of the pedestal jaws, is assumed as a constant and constant it must be.

The increased, and increasing, demands of railroad transportation, in dependable service and economical operation, demand general recognition of the necessity for closer tolerances in both design and practice. The slight increase in initial cost to provide such tolerances is but a small portion of the higher operating expense resulting from a denial of this necessity.

Authors' Closure

The appreciated, well-qualified, and pertinent discussion raises two questions: What are the results of service tests on the Disc-Flo unit, and to what degree can the fundamental Disc-Flo improvements in journal-box and bearing design be applied to existing A.A.R. equipment.

At the presentation of this paper, 40 journal units in test service had accumulated 5,329,206 journal-bearing miles, or an average of 661,151 car-miles per car set, during which changes were made as described, incorporating lateral control, and improvement in disk drive and seal construction.

Since June, 1938, 48 journal units have accumulated 3,220,664 journal-bearing miles, or an average of 402,583 car-miles per car set. All of these journal units are in high-speed passenger service, and have given satisfactory operation with no maintenance other than routine terminal inspection.

A tender set of 6×11-in. journal units is being installed under a heavy high-speed passenger-locomotive tank and a car set of $3^3/_4 \times 7$-in. units has just been installed in a modern high-speed streetcar, from which additional service data will soon be available.

Intensive study has been given to the application of the Disc-Flo improvements to the Standard A.A.R. journal box and bearing assembly.

A dust guard incorporating the design fundamentals of the Disc-Flo seal has been developed for the A.A.R. box. It consists of a resilient flexible sealing member sandwiched between two aluminum-alloy plates, one a seal plate, the other a follower plate. These plates are provided with bores of different diameters; the seal plate adjacent to the wheel fits snugly on the dust-guard seat of the axle, and the follower plate, having a larger bore, permits the flexible member to form an annular wiping flange which effectively seals the fit around the axle. The seal plate, due to its close fit on the axle, guides the seal assembly in following the movement of the axle in reference to the box, as the dimensions of the seal unit are such as to afford its unrestricted movement in a standard dust-guard well. The spring action of the resilient member holds the aluminum plates against the inside and outside walls of the journal-box cavity with sufficient pressure to seal against entrance of dirt and yet not interfere with the movement of the seal assembly.

Over 700 of this type of dust guard in its various stages of development are in service, and the oldest installation is still giving good service, having made a mileage of 260,000 in high-speed passenger service as of December 1, 1938.

The system of lateral control, standard in Disc-Flo design, can be applied without modification to A.A.R. axle, box, and bearing assemblies. By establishment of close tolerances on new and reworked axles, on bearings, boxes, wedges, and truck parts, and modification of dimensions of lateral-component lugs and collar of journal bearing, the evils of excessive lateral play can be overcome.

The degree to which it is desired to carry the bearing improvement as dictated by cost and departure from existing standards, requires a decision as to whether the standard steeple-backed A.A.R. journal bearing should have its collar and thrust-lug dimensions modified, or the standard wedge and bearing should be abandoned in favor of the flat-back lateral-control bearing with its modified wedge.

By modification of the collar and thrust-lug dimensions of the standard A.A.R. bearing, the desired reduction in axle free lateral play can be secured, but to stabilize the respective running positions of the axle and the bearing and to increase the thrust capacity of the bearing the flat-back type has to be resorted to.

In answer to Mr. Warden's doubt as to the ability of the flat-back bearing with extended sides to overcome tilting, it should be pointed out that with this combination, using a flat wedge, there is no inclined plane formed by the wedge for the bearing to climb, when horizontal impact is imposed on the axle. Further the force of the horizontal axle impact against the more nearly vertical extended side of the bearing exerts relatively small lifting force on the bearing and this has to be in excess of the static vertical load before there is any raising of the bearing.

Over 4600 A.A.R. type bearings modified for lateral control have been put in service with results that justify the foregoing statements.

One hundred thirty-two flat-back lateral-control bearings have been applied in heavy locomotive-tender service, and on high-speed passenger cars, with a single brake shoe per wheel. Under this severe service, the life of the journal bearings has increased beyond all expectation.

Problems in Modern Deep-Well Pumping[1]

Carlton W. Dawson.[2] The author has presented a clean-cut mathematical analysis of deep-well pumping problems, and engineers who deal directly with these problems may learn much from his careful study.

Prior to this time the importance of the step-tapered rod string (the closest approach to the rod string of uniform stress) has been recognized and used almost exclusively in sucker-rod pumping at depths greater than 7000 ft. Reasonable rod service has been achieved, but production rates have been well below the maximum rates given by Equation [18] of the paper.

The author has analyzed plunger overtravel as a function only of the accelerations at the ends of the stroke. Assuming harmonic motion, the polish-rod-stroke correction factor becomes $F = 1 + [1.93\,(LN)^2/10^5]$. However, to give results in closer agreement with those obtained by field tests with the Gilbert-Sargent bottom-hole dynagraph, it is suggested that the pitman-crank-length acceleration factor be applied. With a ratio of four, the foregoing formula becomes $F = 1 + [2.4\,(LN)^2/10^5]$. This correction assumes the magnitude of acceleration at both ends of the stroke to be equal and opposite in sign. Actually, in the conventional pumping unit, the acceleration at the bottom of the stroke is due to the individual crank and pitman accelerations being additive when at 180 deg. Conversely, when at the top of the stroke the pitman overlies the crank and the acceleration of the pitman is subtracted from that of the crank. The sum of top and bottom acceleration is approximately equal to the sum if the motion be actually harmonic. The significance of this is that the first correction is the only one theoretically justified.

But, because calculations based on this formula do not agree with actual performance, it is obvious that a more involved treatment is justified. Various applications of Timoshenko's work on vibrations have been attempted; notably those of R. W. Rieniets[3] and Kendrick and Cornelius.[4] This approach gives promise of being much more satisfactory, although a great deal more correlation of field data is required to provide the essential support.

In the summary of the paper, the author states that pneumatic counterbalances are adapted to heavy loads and will give lower peak loads than beam or crank balances. This is true, but is due not to the type of balance but to the effect of the accelerations. The point of attachment of the pitman and crank on air-balanced units is between the samson post and polish rod, whereas the samson post of the conventional units is between the pitman and polish rod. As pointed out previously when the pitman overlies the crank the accelerations of each are of opposite sign, and when they are extended at 180 deg they are of the same sign. This means that the air-balanced units have greater rod acceleration at the top of the stroke where the effect is to reduce the rod load. The conventional unit has the greater acceleration at the bottom of the stroke where the effect is to increase the rod load.

Author's Closure

Mr. Dawson's criticism of the relation $F = 1 + [2.4(LN)^2/10^5]$ is well founded. This seems to fit field results more closely than the previously used relation, but its use must be based on its value as an empirical relation rather than a fundamental of crank motion.

The air balance whether directly applied as in the Sullivan head or in combination with a crank motion tends to smooth out the load measured at the polished rod. It is true that the point of attachment of the pitman is a factor, but the absence of the large mass of the counterbalance is thought by the writer to be of greater importance.

[1] Published as paper PME-60-2, by C. J. Coberly, in the October, 1938, issue of the A.S.M.E. Transactions, vol. 60, p. 561.

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[3] "Plunger Travel of Oil-Well Pumps," by R. W. Rieniets, American Petroleum Institute, Drilling and Production Practice, 1937, p. 159.

[4] "The Sucker-Rod Pump as a Problem in Elasticity," by J. F. Kendrick and P. D. Cornelius, Transactions A.I.M.E., Petroleum Division, vol. 123, 1937, p. 15.