Transactions

of the

A.S.M.E.

OCTOBER, 1940

VOL. 62, NO. 7

Transactions

of The American Society of Mechanical Engineers

Published on the tenth of every month, except March, June, September, and December

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Published monthly by The American Society of Mechanical Engineers. Publication office at 20th and Northampton Streets, Easton, Pa. The editorial department located at the headquarters of the Society, 29 West Thirty-Ninth Street, New York, N. Y. Cable address, "Dynamic," New York. Price $1.50 a copy, $12.00 a year; to members, $1.00 a copy, $7.50 a year. Changes of address must be received at Society headquarters two weeks before they are to be effective on the mailing list. Please send old as well as new address.... By-Law: The Society shall not be responsible for statements or opinions advanced in papers or .... printed in its publications (B13, Par. 4).... Entered as second-class matter March 2, 1928, at the Post Office at Easton, Pa., under the Act of August 24, 1912....

Successful Mechanical Atomization of Fuel Oil Heavier Than Bunker C

By G. G. MARTINSON,[1] MARCUS HOOK, PA.

The purpose of this paper is to report the difficulties encountered and their successful solution when burning a heavy-petroleum cracking-still residue as a fuel in combination pulverized-coal and fuel-oil burners for the boilers of the American Viscose Company powerhouse at its Marcus Hook plant. Though the problems involved were comparatively simple, the development of a burner arrangement to achieve a solution and the results obtained may be of value to the designers and operators of boilers who are in search of both flexible and efficient combination firing equipment for pulverized coal and fuel oil in boiler plants.

EQUIPMENT of the boiler plant of the American Viscose Company at Marcus Hook consists of three boilers of C.E. type fin-tube waterwall-furnace construction, each having a maximum steam output of 150,000 lb per hr at 425 psi and 700 F total temperature. Fig. 1 shows a side view in cross section of one of these boilers. The furnaces, of the dry-bottom type, have a volume of 7000 cu ft and a waterwall surface of 2300 sq ft. Steam from the boilers is used to drive three 3000-kw turbogenerators and three 2000-kw turbogenerators. Some of the steam is extracted from the turbines at 175 psi and some at 20 psi for plant process duty.

Each boiler is equipped with four Lo-Pul-Co type C.E. combination burners for oil and pulverized-coal firing, the general arrangement being shown in Fig. 2. When firing pulverized coal, the fuel is mixed with tempered primary air at the pulverizers and delivered through the center pipe of the burner assembly, while secondary air is forced through the air preheater by a forced-draft fan and delivered to the burner assembly outside of the pulverized-coal pipe. Control of the secondary air is accomplished by regulating the register shown in Fig. 2 and the speed of the forced-draft fan. Primary-air fans are direct-connected to the bowl-mill motors and, therefore, when using oil as fuel for these furnaces, no primary air is delivered through the coal pipe, combustion depending entirely upon the mixture of the fuel oil and secondary air supplied in the manner described.

Control of the firing rate of each of these units is completely automatic for both types of firing equipment and, after setting the proper air-fuel ratio for the type of fuel used, the controls maintain this ratio automatically within close limits throughout the range of the controls. An increase in steam demand causes the controls to increase the air supplied to the burners before increasing the fuel supply and, with a decrease in load, the fuel supply is decreased before the air supply is reduced. This cycle of operations is very effective in preventing smoke when changes in load occur.

The proximity of the plants of the American Viscose Company and the Sinclair Refining Company made the direct delivery of heavy fuel oil by pipe line from the latter to the former a rather simple problem. Fig. 3 shows a flow diagram of the manner in which this heavy fuel oil is measured and delivered to the consumer's burners without the heat loss attendant upon other methods of fuel-oil delivery, such as tank-car and barge deliveries. The refinery delivers this heavy fuel oil to the supply tank shown in Fig. 3 at a temperature about equal to the optimum temperature at which the oil is most readily atomized. Therefore, in order to have the fuel oil at the correct temperature for atomization at the burners in the Viscose plant, it is only necessary for the consumer to supply the heat lost in the oil delivery line.

Difficulties Encountered in Oil Firing With Original Equipment

The original oil burners and burner assemblies gave unsatisfactory results because the atomized fuel oil and the air required for combustion were not properly mixed. This condition caused the atomized-oil streams to impinge upon the side-wall tubes of the furnace where the oil distilled and the residue formed a coke deposit which kept building up in size until the weight of the mass of coke overcame its adhesive powers and the entire mass fell onto the screen tubes across the bottom of the furnace. Some of these blocks of coke were heavy enough to bow the floor screen tubes slightly.

Fig. 2 shows the combination burner and air register in cross section with the normal angle of atomized-oil spray from the standard mechanical atomizer and illustrates the relative positions of the air and oil streams. In this illustration the oil burner is shown withdrawn to the greatest possible extent without impingement of the oil on the nozzle of the coal pipe. It can be readily noted that the flame in this position fails to fill the throat and, since the stream of air leaving the secondary-air passage is "barreling," there is little mixing of the air and fuel except at the outside of the oil-flame cone. Therefore, a considerable portion of the air introduced into the furnace in this manner is not efficiently utilized in combustion and, although excess air exists in the furnace, smoke and soot fill the firebox and boiler setting.

Steam atomizing oil burners with the tips drilled to give approximately the same included angle of flame cone as that produced by the mechanical atomizing burner tips were tried and produced only slightly better results than the mechanical atomizers. Experimental work was carried out with steam atomizing oil-burner tips with various drillings; it developed that a tip drilled to give a flame included angle of about 120 deg produced the best results. With this tip, the flame could be pulled back into the throat, without impinging on the coal-pipe nozzle, thereby forcing the air into the flame and resulting in good mixture of the fuel oil and the air required for combustion. The amount of steam consumed by the steam atomizers, while probably in line with the steam consumption of other modern steam atomizing oil burners, was considered to be an unnecessary loss. Therefore, a further solution to the problem, using mechanical atomizing oil-burner tips, was sought.

Design, Installation, and Operation of Fixed Air Register

Since the normal mechanical atomizing oil burner gives a flame cone with an included angle of about 75 to 85 deg, which experience with steam atomizing burner tips had proved to be too

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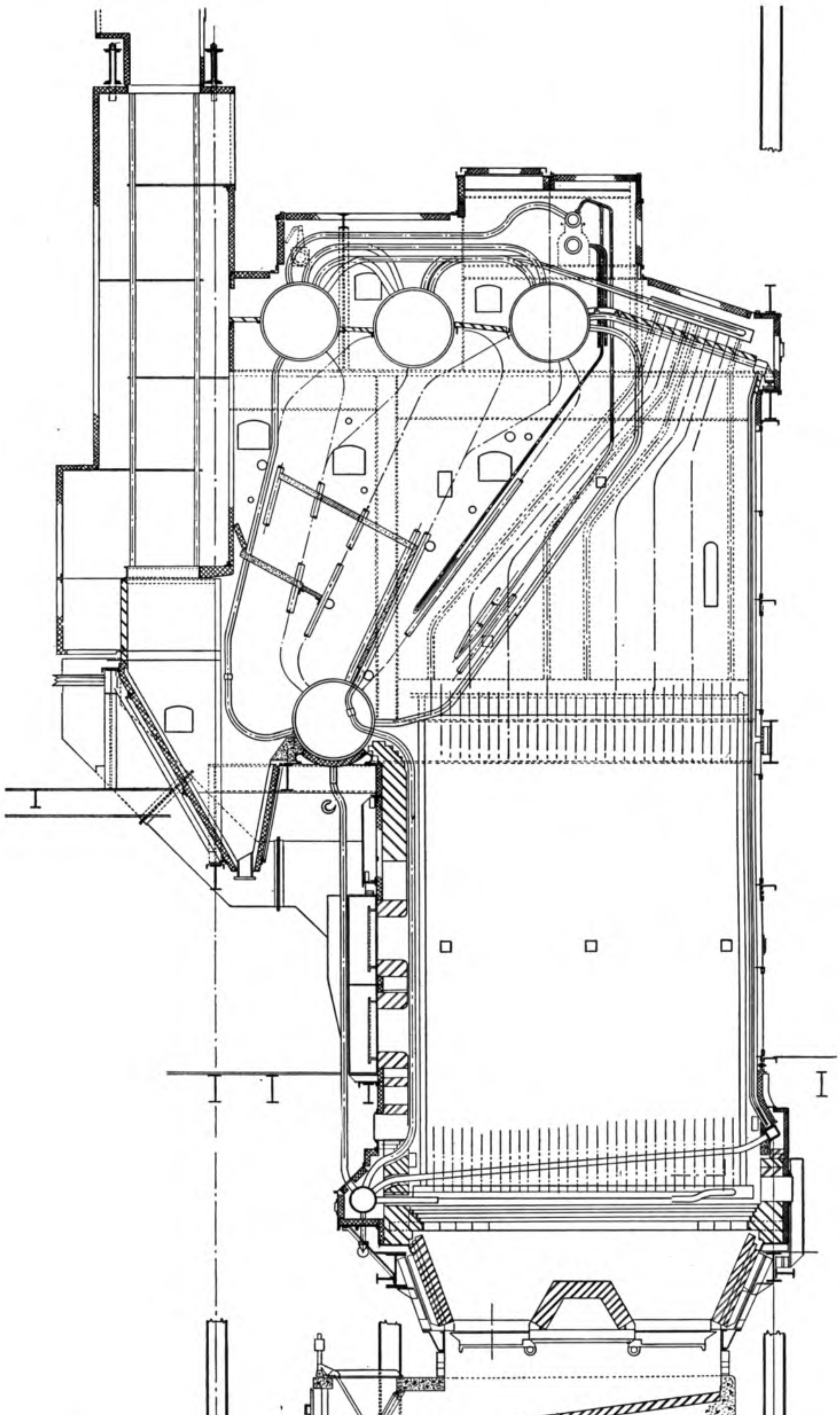
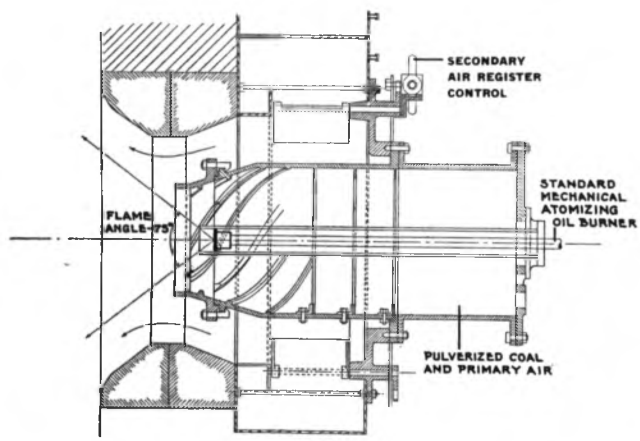
Contributed by the Fuels Division and presented at the Spring Meeting, Worcester, Mass., May 1–3, 1940, of The American Society of Mechanical Engineers.



Courtesy Combustion Engineering Co.

FIG. 1 SIDE VIEW OF MARCUS HOOK PLANT BOILER IN CROSS SECTION narrow, it was deemed advisable to change the manner in which the air was introduced into the furnace in an effort to mix the combustion air properly with the fuel oil. Another important reason for the decision regarding the air introduction was the fact that firing conditions using pulverized coal as the fuel were very good and it was felt that a change in air introduction would have less effect upon coal-burning characteristics than a change in the coal nozzle. The coal nozzle could, of course, be shortened to the extent of permitting the oil burner to be withdrawn far enough to enable its normal flame included angle to fill the throat of the burner. However, this procedure would undoubtedly disrupt the coal-burning characteristics of the burner assembly.

To direct the air stream into the flame produced by the standard mechanical atomizing oil burner, an assembly of 24 fixed



Courtesy, Combustion Engineering Co.

FIG. 2 ORIGINAL BURNER ASSEMBLY WITH STANDARD ATOMIZING OIL BURNER

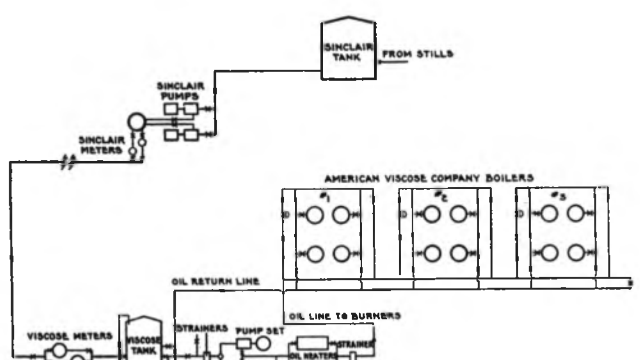


FIG. 3 DIAGRAM OF OIL FLOW FROM REFINERY TO CONSUMER'S BURNERS

vanes was made into a circular register, fitting between the coal nozzle and the refractory throat of the burner assembly, and placed at the furnace inlet of the secondary-air supply. Fig. 4, although it shows a cross section of the burner assembly fitted with a movable-vane register, indicates the position of the registers of both the fixed type and the movable-vane type which was developed later. The vanes of the fixed register were arbitrarily set at an angle of 45 deg to the face of the burner assembly as it was thought that this setting would be sufficiently accurate to determine definitely the feasibility of this method of mixing the air and fuel oil more effectively. The shroud around the outer ends of the vanes was formed to produce a constriction on the air stream by making the outlet diameter smaller than the inlet and, thereby, to assist in obtaining a good mixture of fuel oil and air.

Tests were made, using these fixed-vane registers under actual firing conditions, which indicated that the angle chosen for the vanes had been too great for any normal load on these furnaces. Under light loads, the fire had an excellent appearance, all of the fuel oil being consumed completely within a short distance of the burner tip but, as the load was increased and the fuel-oil and air quantities increased, the flame was torn apart and a "sunflower" appearance of the flame became very noticeable. There was a short solid flame for about 3 ft from the burner tip, then flame propagation was apparently too slow for the speed of the combined air-and-oil-stream mixture and no flame was apparent for the next 4 or 5 ft of travel, after which ignition again took place and combustion was completed.

One of the surprising features of the test was that the flame produced, even though of such unorthodox shape, was remarkably stable and gave no indications of "puffing" or "backfiring." Since the direction and rotation of the air by the fixed-vane register caused the flame to flare out into the sunflower shape, it is

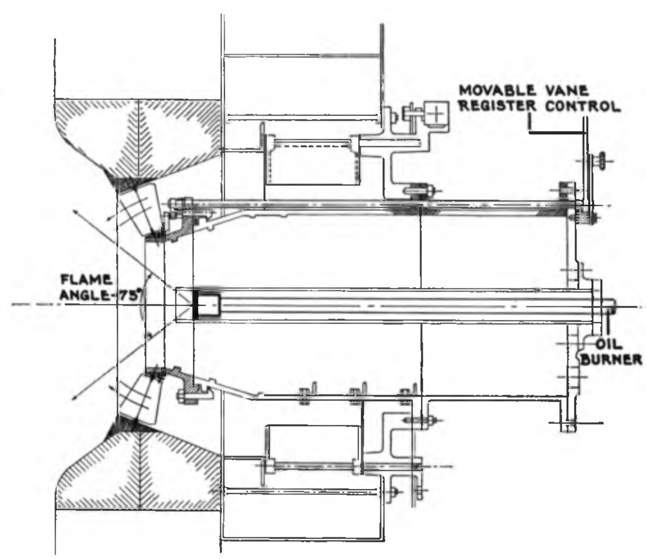


FIG. 4 MOVABLE-VANE AIR REGISTER FITTED TO ORIGINAL BURNER ASSEMBLY

obvious that there was no real length of flame extending into the furnace and most of the heat was concentrated on the front wall.

DEVELOPMENT OF MOVABLE-VANE AIR REGISTER AND ITS OPERATION

The logical procedure from this point was to develop a movable-vane register which could be controlled from the firing floor of the boiler room, varying the position of the vanes with changes in load sufficiently great to require their readjustment. The register shown in Fig. 4 was designed with movable vanes located in about the same position in the burner assembly as the fixed vanes had been in the previous test. Each of the movable vanes was connected to a control ring by a small lug welded to the corner of the vane and riding in a slot in the control ring. This control ring was connected, through suitable linkage, to a rod which extended through the front plate of the burner assembly, terminating in a handle by which the vanes could be adjusted while the burner was in operation and could also be locked in a given position.

After the movable-vane air registers had been constructed and installed, a test was made to determine their characteristics using oil as the fuel and varying the load on the boiler within limits allowed by plant operations. Some difficulties were encountered in the operation of the movable vanes of the register, chiefly due to the expansion of the 18-8 chrome-nickel alloy from which the vanes had been made. Greater clearances were given to the vanes and their bearing supports and no further problems arose from that source.

Further tests indicated that a position of the vanes, suitable for normal load swings, was about $^2/_3$ open at about 80 per cent maximum output of the boiler. Decreasing the load on the boiler more than approximately 15 per cent required the closing of the vanes of the register proportionately and vice versa. Firing conditions on oil were considered satisfactory as combustion was complete and the flame shape and length could be controlled by the adjustment of the movable-vane air registers.

The characteristics of the revamped burner assembly had not been determined with pulverized coal and tests were therefore started to investigate the firing conditions with that type of fuel. As previously mentioned, the secondary-air stream from the original burner assembly entered the furnace in approximately a hollow rotating cylinder form or "barrel." This air-stream shape had been disrupted and changed in both direction and shape by the addition of the movable-vane air register at the secondary-air outlet. Tests made with coal as the fuel and the movable-vane register in the air stream were generally unsatisfactory as the register in the wide-open position seemed to take any rotational effect out of the secondary air that had been imparted to it by the register of the original equipment, causing an unstable flame condition. If the movable vanes of the added air register were partially closed, the flame was also unstable and ignition was difficult to maintain.

A proposal was made that deflectors be installed in the pulverized-coal pipe to direct the pulverized coal and primary-air stream toward the secondary-air stream in a manner similar to the shape of the atomized-oil stream. It is believed that this would have overcome the unsatisfactory coal-burning characteristics of the movable-vane register, but the idea was tabled in favor of further consideration of the original equipment using mechanical atomizing oil burners of another make in place of the burners originally supplied.

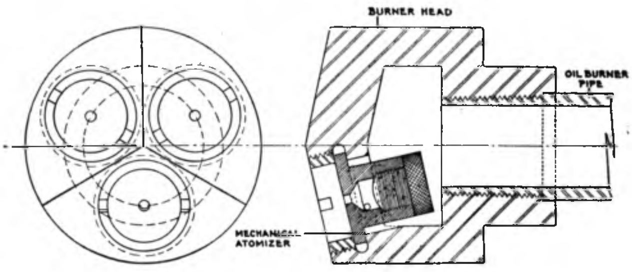
Sinclair engineers had definitely established the fact that the heavy fuel oil supplied by the refinery could be efficiently burned as fuel in the Viscose plant, even though the furnaces were completely water-cooled and the resultant furnace temperatures were comparatively low.

In seeking an adequate solution, another manufacturer was permitted to demonstrate a different type of mechanical atomizer. This burner manufacturer had had considerable experience with mechanical atomization of fuel oil in marine boiler furnaces and it was felt that engineers of the company might suggest a solution to the problem based on previous experience. Tests were made under the direction of two engineers, using this company's mechanical atomizers in the original burner assemblies, both with and without additional deflectors placed on the oil-burner guide pipes. These deflectors could be moved in or out along the oil-burner guide pipes. However, changing the position of the deflectors seemed to have but little effect on the fires, since no primary air passes through the coal pipes on oil firing and the deflectors could have no effect on the secondary-air streams.

Because of the fact that the mechanical atomizers used were more or less of the conventional type with the usual included angle of flame cone of about 75 deg, the results obtained from tests of these burners were similar to those obtained on the original mechanical atomizers. When using the atomizers of the second manufacturer, it was impossible to get more than a slight mix between the fuel oil and air and, consequently, a very smoky unsatisfactory condition existed in the furnace. The basic problem of mixing the air from the burner register, as installed originally, with the fuel-oil streams from the mechanical atomizing burner tips was still unsolved.

Wide-Angle Tri-Tip Mechanical Atomizer Successful

While the foregoing tests were being made, one of the Sinclair engineers had been searching for a mechanical atomizer which would produce a wide angle of spray. A mechanical atomizing burner tip was located which had been developed for steel-mill purposes. This tip consisted of three atomizers set in a single burner head in the manner shown in Fig. 5. The included angle of flame cone that would be produced by this arrangement was apparently about 120 deg. Judging from previous satisfactory results obtained with wide-angle steam atomizing tips, it was assumed that the wide-angle mechanical atomizer would be successful. Four of these burner heads were immediately ordered, each containing three atomizing tips of a size estimated as correct for the normal load conditions.



Courtesy Coen Burner Co.

Fig. 5 Tri-Tip Mechanical Atomizing Oil-Burner Head

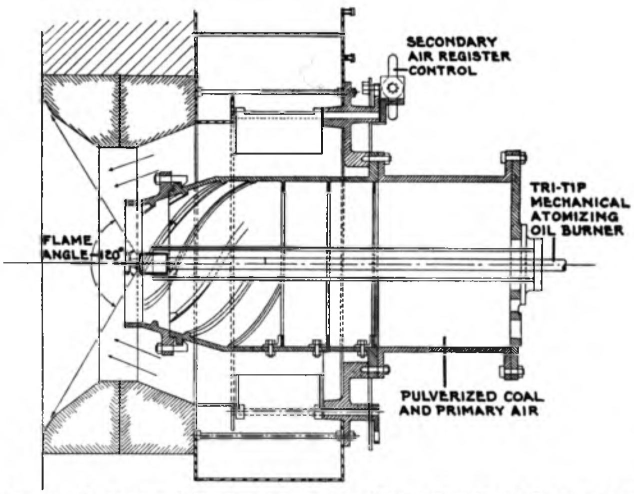


Fig. 6 Original Burner Assembly With Tri-Tip Mechanical Atomizing Oil Burner

Preliminary tests of these burners upon their arrival indicated that they would be entirely satisfactory, except for size, as the included angle of spray was such as to produce the proper air-fuel-oil mixture. The atomizers of the original installation were too large for the load requirements, which resulted in a lower pressure on the fuel oil to the burners than that required to give the best atomization. However, the tests that had been made showed sufficient promise to warrant the purchase of additional burner heads of this type to complete the installation on all of the boilers.

Atomizers of several sizes were ordered, the range in size making it possible to have all four burners of the boiler furnace in operation producing a balanced firing condition and an even distribution of heat in the furnace. Normal changes in load are easily taken care of by the range of the oil pressure to the burner, while the large changes, such as occasionally occur due to starting up or shutting down portions of the plant, can be taken care of in a few minutes by changing the atomizer tips.

In the operation of these burner tips, flame propagation begins within 1 or 2 in. of each of the atomizers but, 6 or 8 in. from the tip, the three streams lose their identity and become one large flame issuing from the burner throat. Because of the wide angle of the combined atomized-oil sprays from these burner tips, the burner can be withdrawn to a point where the fire completely fills the throat without danger of impingement of the oil spray on the coal nozzle of the burner assembly. Thus, the atomized-oil spray is forced into the secondary-air stream, as shown in Fig. 6, and complete combustion takes place with little excess air. Firing conditions are good and the complete operation of these furnaces on heavy fuel oil is entirely satisfactory.

These furnaces have been operated for more than 15 months using heavy oil as fuel, and the costs of steam and power produced during this operating period compare very favorably with those obtained while burning pulverized coal. Additional advantages, not included in the cost statement but important to the operator, are the ease of handling and over-all plant cleanliness of the oil-firing operation.

Automatic Control Satisfactory for Heavy Oil

Very few problems have arisen on the control system used in connection with the firing of heavy fuel oil. The controls vary the amount of fuel oil and air to the burners in response to load changes on the boiler, maintaining the fuel-oil-air ratio set on the master control throughout the range of the control. The fuel-air ratio is manually set for the best combustion on any particular load and the proper ratio is then maintained by the control. A pilot or measuring valve in the fuel-oil line to the boiler burners actuates suitable equipment to damp the action of the control valve in the same line and, thereby, smooths out the flow of oil to the burners. This pilot valve as originally installed was of the piston-type and the heavy-oil fuel had a tendency to stick the piston, making the control sluggish. To remedy this condition, a diaphragm-type valve was substituted.

Physical Properties of Fuel Oil

The heavy fuel oil mentioned in this paper is a residuum product of a petroleum-refinery-cracking process and, as such, the characteristics of the oil may vary slightly from time to time with changes in operation. Representative characteristics of this heavy fuel oil may be listed as follows:

Specific gravity	1.014	Flash point, F	250
B.S.&W., centrifuge, per cent	0.50	Sediment, per cent	0.05
Viscosity, Furol at 122 F	800–1400	Weight, lb per gal	8.45
Pour point, F	80	Btu per lb	18300

A temperature of approximately 275 F has been found satisfactory to reduce the viscosity of the oil to about 20 Furol, or 150 S.S.U. At this viscosity, the oil is readily atomized and its characteristics are such that the viscosity at the firing temperature is about the same, whether the Furol viscosity at 122 F is 800 or 1400. The carbon-hydrogen ratio is greater for the heavy fuel oil than it is for the lighter oils, consequently, the advantage of the heavy fuel lies not only in the additional heat obtained from the carbon but also in the proportionate reduction of loss from the formation of water vapor.

Discussion of Conditions and Results

In general, combination burner assemblies are of incorrect design for the efficient combustion of all the fuels included in the combination. The apparent practice, in the East, has been to design for pulverized-coal firing and to fit in accommodations for some other fuel, such as oil, in the easiest manner possible and with little regard for resultant operating conditions. The true combination burner assembly should handle all fuels intended for it in an equally efficient manner. Such an assembly cannot be designed without a complete knowledge of the characteristics of all the fuels and burner equipment proposed for a particular installation. Since this information may not be available to the designer and since there are many installations of improperly designed combination burner assemblies in operation at the present time, it would be advisable, where one of the fuels used is oil, for the designer or operator to investigate the possible advantages in using a multiple-tip mechanical atomizing oil burner in his original burner assembly.

The number of atomizers per burner head is not limited to three, nor is the included angle of spray from the tip limited to 120 deg. These conditions were found satisfactory for the installation reported in this paper, but another installation might require five or six atomizers, or more, per burner head and a much greater included angle of spray than the 120 deg used in the Viscose installation. It would appear possible, where faulty air distribution around the burner is inherent in the burner assembly, to improve over-all combustion by the selection of proper-sized atomizers and face angles to alleviate this condition to a considerable extent.

Satisfactory combustion of heavy fuel oils, using mechanical atomizing oil burners, can be readily accomplished when the proper equipment is available for heating, atomizing, and mixing the oil and combustion air. The additional cost of this equipment over that required for steam atomizing oil burners can usually be justified on steam saving alone.

Conclusion

The results obtained in this investigation and herein reported have definitely shown that it is not only possible, but practical and economical, to burn heavy petroleum oils in properly designed combination coal-and-oil burners, using mechanical atomization under automatic control, in completely water-cooled furnaces.

Acknowledgment

The advice of O. F. Campbell, R. J. Self, and E. W. Griscom of the Sinclair Refining Company is acknowledged in the successful completion of the investigation and the preparation of this paper. The author wishes to acknowledge, also, the willing cooperation of the American Viscose Company, the Combustion Engineering Company, the Coen Burner Company, and others for their assistance.

Discussion

J. M. Goldstein.[2] Briefly, the paper relates to the operating difficulties encountered in an oil-burning power plant using fuel oil heavier than bunker C. Here particular difficulty was experienced in the building up of coke deposits on the side-wall tubes of the furnace, as after a time these coke deposits fell to the bottom screen tubes. After experimenting with different burner tips and various methods of air introduction, the tri-tip head was used successfully to eliminate the coke deposits building up on the side-wall tubes.

In analyzing the type of oil spray that is obtained with the tri-tip head, it appears that the outside cone consists of a much finer spray than that on the inside of the cone; i.e., the oil particles nearest the center of the head from each tip will meet and the resultant oil particle will become larger and more difficult to

[2] Combustion Service Company, New York, N. Y.

burn completely. Close observation will show decidedly black streaks of unburned oil near the center of the oil fire, with the result that carbon deposits will accumulate on the upper tubes, and there will be excess heat losses due to unburned gaseous hydrocarbons. However, finer atomization on the outside of the flame cone will aid combustion nearest the side-wall tubes, and thus eliminate the difficulty due to coke deposits building up on these tubes.

The use of a specially designed atomizer head to produce a fine oil spray of the desired angle represents, however, only one important factor in obtaining clean, efficient combustion. In order to obtain satisfactory operating results in the burning of heavy fuel oil with mechanical atomizers, other important factors in addition to atomization are oil-air intermixture, combustion process, and combustion atmosphere.

In breaking up the oil into a very fine spray with the mechanical atomizer, the control of the process can be regulated to some extent by adjustment of the oil pressure and temperature. It is desirable that the oil particle, when projected in suspension, should not come in contact with either solid surfaces or other oil particles. However, in the case of some heavier bunker oils, atomization will be poor in spite of the fact that oil temperatures and pressures are adjusted over the entire operating range as recommended for the fuel and oil burner. Inherent characteristics of the fuel itself will affect the quality of atomization. Heavy bunker oils, which are residual fuels, are subject to change in content and characteristics from time to time, and require constant attention to maintain best operation.

A proper oil-air intermixture depends upon the quantity, direction, and velocity of the air supplied in relation to the oil spray. Each oil particle should be completely surrounded by air to promote oxidation of the hydrocarbons. Regulation of the oxidation process will control combustion so that it is neither too rapid nor too slow, thereby greatly diminishing the "cracking" or thermal decomposition of the oil particle.

When combustion takes place due to the oil-air intermixture, a combustion atmosphere is created in the furnace. Since the air used for combustion is subject to change due to temperature, humidity, and contamination, it follows that proper combustion is also a function of a proper quality of air supply.

The common practice of sampling fuel oils in a laboratory for analysis and burning characteristics has proved inadequate. For example, good ignition characteristics and balanced distillation are important factors affecting combustion. Then again, fuel oil, after leaving the refinery, is subject to contamination in transportation, storage, and handling, which also affects atomization and combustion.

Again referring to the paper, it is shown that the fuel oil is transported by pipe line from the refinery to the power plant with small loss in temperature; also that the boilers are of waterwall-furnace construction. These conditions are unusually favorable for oil burning, and are the exception rather than the rule. For instance, in most installations fuel oil is delivered either in tank trucks or barges and, as mentioned, becomes subject to contamination which in turn increases the difficulties in burning. In many cases, refractory material is used in combustion chambers, with and without facilities for cooling. The presence of noncombustibles in the oil, such as natural salts, chemicals used in the refinery operations, scale, and dirt will cause rapid deterioration of the refractory materials, particularly at high combustion-chamber temperature.

The mechanical atomization of heavy bunker oils is a problem which has confronted engineers in charge of every type of installation used for heating and processing. In many plants, high-pressure steam for steam atomization is not available, which automatically eliminates this method of solving combustion problems.

Methods must be provided to control within limits the variable factors which affect atomization and combustion. Recent tests with the use of a chemical oil conditioner to improve the atomization and combustion of light and heavy fuel oils have resulted in the solution of many common difficulties encountered in oil burning.

A recent troublesome installation, similar to the case cited, where carbon deposits formed in the furnace, was in an apartment building in New York City, where a rotary-type oil burner using bunker C oil had been installed in a low-pressure fire-tube boiler several years ago. The combustion chamber was built of standard firebrick, and was too short and too narrow for the size of the boiler. A dutch oven was recommended but not installed.

The oil fire impinged somewhat on the side walls but much more on the rear wall of the combustion chamber. During the several years that the burner was in operation, the quantity of carbon deposits which accumulated on the walls of the chamber and in the fire-tubes over a 30-day period was not excessive or bothersome. However, by the fall of 1939, the carbon deposits on the rear wall of the chamber became excessive and required removal every two hours when the burner was in operation. Oil temperature and pressure were changed, primary- and secondary-air adjustments were regulated, and even the oil supplier was changed, but the carbon deposits kept building up. The oil-burning system was reset for normal operation, using the same bunker oil, and "chemical conditioning" was applied. Now for the last several months, in spite of continued flame impingement on the walls, carbon deposits have been eliminated, and trouble-free, efficient operation has been maintained.

Another interesting case is the conversion of several coal-burning boilers to oil burning by a public-utility company. The bunker oil used is a by-product of the gas-making division of this company, and a standard-make mechanical-atomizing oil-burning system was installed and operated in accordance with the manufacturer's instructions. Difficulties encountered were high furnace temperature and melting of the refractory material in the combustion chamber at normal overloads. There was no flame impingement, fires were sparky, and there were complaints of smoke nuisance when a boiler was put on the line. Changes were made in the type of burner tips, design of the combustion chamber, air delivery, and quality of firebrick, but operation was only slightly improved. As a further remedy, this heavy-residuum bunker oil is now treated so that its specifications are equivalent to or better than a commercial bunker C fuel oil. Further experimentation is being carried on by the company engineers in cooperation with the equipment engineers to burn this residuum by-product successfully, without the necessity of this expensive treatment.

In view of the fact that in recent years revolutionary new catalytic processes have been developed in petroleum refining, and since bunker oils are thereby subject to change, it may prove difficult to design a mechanical atomizer which will function efficiently with all grades of bunker oils under all conditions. When mechanical changes fail to solve oil-burning problems, the chemical-conditioning method may prove economically sound and advantageous. The consumer of fuel oil and his staff should seek the cooperation of the oil supplier and equipment manufacturer, and in addition, the impartial advice of the combustion specialist.

R. C. Vroom.[3] The paper emphasizes the desirability of installing suitable combined burners initially, when it is possible that future requirements may make it necessary to burn a liquid or gaseous fuel instead of a pulverized fuel or vice versa. The writer feels, however, that the paper may have created an impres-

[3] Chief Engineer, Peabody Engineering Corporation. Mem. A.S.M.E.

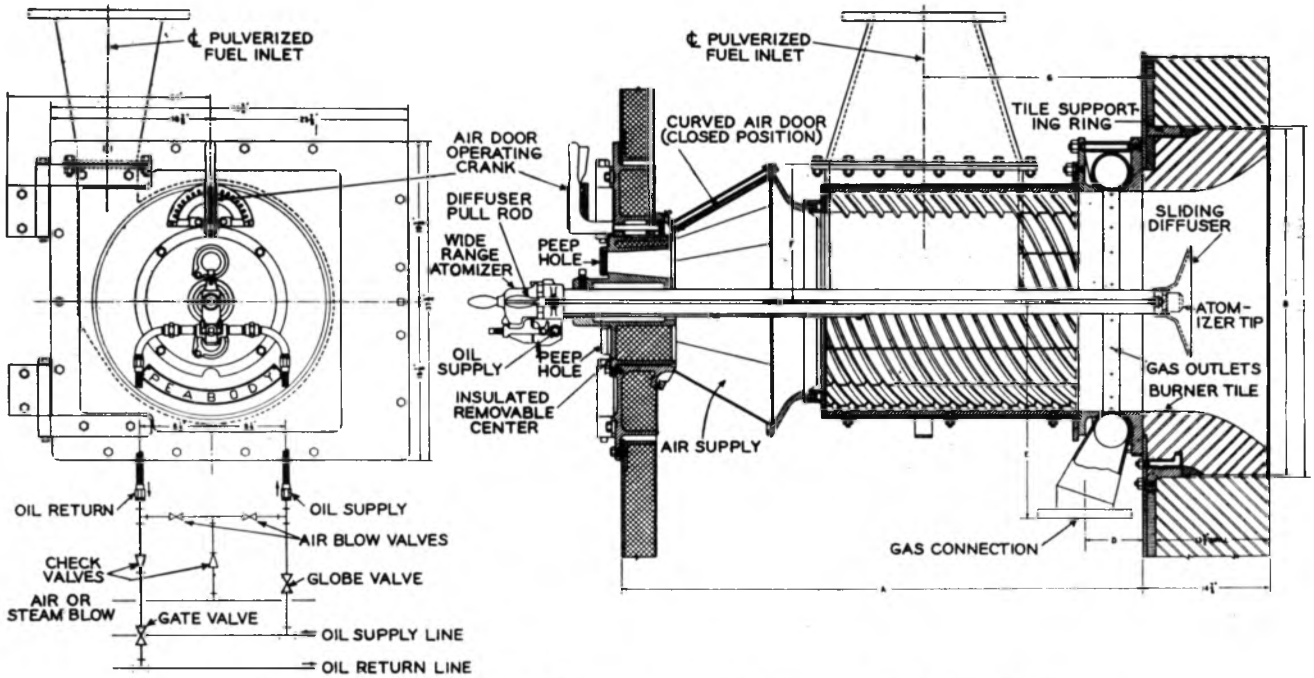


FIG. 7 COMBINATION BURNER DESIGNED TO BURN PULVERIZED FUEL, GAS, AND OIL

sion which surely was not intended, i.e., prior to the experimental program which the author describes, there were not available suitable combined burners, nor atomizers, of the mechanical type which could handle liquid fuels heavier than bunker C.

For nearly 15 years the company with which the writer is connected has been building combined burners for liquid, gaseous, and pulverized fuels varying in capacity from 15 to 150 million Btu per hr liberation each. They have been described in papers presented before the A.S.M.E. and a typical burner is illustrated in Fig. 7 of this discussion. These burners are designed to meet and do meet combustion guarantees which are in every respect competitive, and no difficulty is experienced in burning individually any of the fuels which the burner is designed to handle, nor in changing from one fuel to another without loss in steaming capacity. They are successfully firing, in many of the well-known steam plants in this country, in units varying in capacity from 20,000 to 500,000 lb of steam per hr each.

In connection with mechanical atomizers for heavy liquid fuels, the writer spent about 6 weeks in Louisiana 10 years ago running tests with such atomizers on all sorts of refinery-waste fuels. These included not only neutral fuels but acid and alkaline sludges, which are far from homogeneous, may contain various quantities of water, and tend to precipitate solids. As a result of these tests, mechanical atomizing liquid-fuel burners were installed in a new utility plant which was to use the waste fuels purchased from the refinery and in turn supply the refinery with process steam. In addition to the waste liquid fuels, it was necessary to burn petroleum coke from the refinery and also natural gas to make up the deficiency between the available waste fuels and the total fuel requirements.

The initial installation comprised four boilers, each to produce 350,000 lb of steam at 600 psi and 750 F temperature. Two of these boilers were equipped with 18 burners each, for liquid and gaseous fuels, and two with 15 such burners, and also two burners for handling the pulverized coke. More recent extensions have employed a smaller number of higher-capacity burners per boiler. There are several other plants which over a number of years have successfully employed mechanical atomizers for firing high-capacity boilers with such fuels, and also plants in which heavy residual fuels and tars with melting points of 300 F or over are being handled. In addition, of course, there are hundreds of combined burners in use where the liquid fuel is ordinarily bunker C.

T. H. WALKER.[4] The design of combination burners to fire pulverized coal and oil, or coal, oil, and gas has been given considerable study, since economic conditions or character of plant load often justify their use. As pointed out in this paper, such burners should handle all fuels equally well, i.e., without operating difficulties or smoke nuisance, and with combustion efficiencies corresponding to best practice for each fuel. Usually the change from one fuel to another must be made without shutting down the unit and sometimes two fuels must be burned together; therefore, no device can be added which will improve operation with one fuel at the expense of that with another.

It has been and to some extent still is the practice with oil-and-gas firing to use a number of small burners. This assists in the utilization of all the furnace volume, reduces the flame length, and to some extent simplifies the problem of introducing the correct amount of air at the proper place, since the flames tend to be mutually sustaining and the burners are close enough to each other so that a slight excess of air from one burner may tend to compensate for a deficiency in another. Also, wide-range operation can be obtained by shutting down some burners and operating others at approximately their design velocities.

Pulverized-fuel burners for the same steam delivery must be fewer and larger because no satisfactory means has yet been devised for splitting a coal-air mixture into many small streams at the same pressure and air-coal ratio; the use of many small mills would be complicated and expensive. Thus the individual burners must be capable of handling a wider load range. Thorough mixing of fuel and air as they enter the furnace, a minimum of excess air, and stability of ignition are essential. In combination burners these objectives must be accomplished with fuels of quite different physical characteristics, since the volume of coal-air mixture is greater than that of either oil or gas. The coal is

[4] Combustion Engineering Company, Inc.

carried to the burner by a quantity of air which often is a sizable proportion of that required for combustion and, hence, requires less secondary air. In burners of the type under discussion where furnace turbulence must be obtained by the use of a turbulent air stream, the fuels must be introduced concentrically, and the necessity for a rather large coal-air nozzle, as compared with the size of oil or gas burners, moves the latter further away from this turbulent air stream than would be the case in similar burners designed for oil or gas only.

With these considerations in mind, the solution which the author describes is interesting. The advantage of the tri-tip burner lies in its ability to provide a wide-angle spray which can be directed into the secondary-air stream, thus accomplishing more thorough mixing of fuel and air than the conventional single-jet atomizer.

Author's Closure

Mr. Goldstein's observations regarding the oil spray obtained with the tri-tip burner head apparently did not take into account the action of the turbulent-combustion air stream on the oil spray from this type burner head. An intimate mix is obtained between combustion air and the atomized fuel oil, as reported in the paper, and observation of the fires, from any of the furnace observation ports, reveals no black streaks of unburned oil in the center, or at any other portion of the flame. These boilers were recently shut down for internal inspection and no appreciable deposits were found in the furnaces or flue gas passages. This would appear to be a further indication of complete consumption of the carbon in the fuel oil. This writer's discussion also states that the waterwall-furnace construction of these boilers is unusually favorable for oil burning. It is felt that most authorities agree that fuel oil requires a higher furnace temperature for good combustion conditions than other fuels, i.e., pulverized coal or gas. Furthermore, furnaces designed primarily for fuel-oil firing usually contain more refractory, to maintain a higher temperature, than those furnaces designed primarily for pulverized coal. The furnace walls of the boilers which were described in the paper are almost completely covered with water-tubes and, due to the amount of heat absorbed by these tubes, the furnace temperatures are considered low.

Mr. Goldstein reports a few interesting cases of oil-firing difficulties but they seem to be concerned with lighter fuel oils and are therefore hardly comparable to the installation described in the paper.

Mr. Vroom's discussion is of great interest to the author. Combination burners for liquid, gaseous, and pulverized fuels, as he points out, have been on the market for many years. A great many of these combination burners, however, have left much to be desired when operating on one or more of the fuels for which they were designed. Revision of such an installation, to correct the undesirable condition, becomes a problem of that installation only; the solution found satisfactory in one instance might or might not be successful if applied in another case. There are combination burners in operation with undesirable firing conditions recognized by the operators but accepted by them as unalterable. Consequently, no effort is made to correct the difficulties.

In his discussion, Mr. Walker emphasizes many of the points which the author sought to bring out in the paper. The observations regarding general practice on burner spacing and placement, with his explanation of the reasons for such spacing and placement, tend to clarify the statement of the problem involved in the original installation.

The author wishes to thank all the discussers of the paper for their contributions to the general subject.

The Analogy Between Fluid Friction and Heat Transfer

Discussion of Paper[1] by Th. von Kármán

By BORIS A. BAKHMETEFF,[2] NEW YORK, N. Y.

IN HIS paper[1] Dr. von Kármán has focused light on a fundamental precept, the import of which lately may have become somewhat eclipsed.

The heat-transfer analogy was originally offered by Reynolds as a fascinating hypothesis to be probed and substantiated by observation and experiment. Quantitatively, the interpretation from the very first was in terms of momentum, the heat exchange due to turbulent convection being appraised at

$$q = \frac{c\tau\Theta}{U} \qquad [a]$$

This basic relation, equivalent to Equation [19][3] and leading directly to von Kármán's

$$C_H = C_f/2 \qquad [16]$$

was accepted in all subsequent interpretations and improvements without scrutiny, even if certain physical aspects cast doubt as to the validity of expressing the Reynolds analogy in that particular form. Indeed as Equation [a] applies to the heat exchange in the turbulent zone only, the symbol $\Theta = \Theta_\delta - \Theta_{fl}$ should signify the difference between the temperature of the fluid Θ_{fl} and a certain temperature Θ_δ at the boundary between the turbulent core and the laminar region. Accordingly Θ in all cases must be less than the full temperature difference $\Theta_0 = \Theta_w - \Theta_{fl}$ between the wall and the fluid. In fact one may properly introduce

$$\eta = \frac{\Theta_\delta - \Theta_{fl}}{\Theta_w - \Theta_{fl}} = \frac{\Theta}{\Theta_0} \qquad [b]$$

as a "reduction coefficient," featuring the "insulating effect" of the laminar zone.

The practical meaning is, that if Equation [a] were to express correctly the physical facts relating to turbulent heat transfer, as for example in a pipe, then the quantities of heat observed experimentally should under all circumstances be below the figures computed from Equation [a] with $\Theta = \Theta_0$. In other words, observed values should comply with

$$q = \eta \frac{c\tau\Theta_0}{U} \qquad [c]$$

where η is < 1.

Such, however, is not the case. Since the early experiments by Stanton and Pannell,[4] it has been known that, for air and gases for which the Prandtl number $\sigma = \mu\, c/k$ is < 1, measured heat values are in excess of q, computed from Equation [a] with $\Theta = \Theta_0$. Accordingly, the over-all experimental values of the heat-transfer number C_H in Equation [5] exceed $C_f/2$. Mathematically, this discrepancy has been repaired by G. I. Taylor, Prandtl, and von Kármán, who complemented the original Reynolds disclosure by accounting supposedly for the effect of the near-to-the-wall laminar film, across which the transfer of heat is by conduction. To elucidate, one may refer to the prototype Taylor Equation [21], written in the form

$$\left.\begin{aligned} q &= \frac{c\tau\Theta_0}{U}\left[\frac{1}{1 + U_\delta/U\ (\sigma - 1)}\right] \\ C_H &= C_f/2\left[\frac{1}{1 + U_\delta/U\ (\sigma - 1)}\right] \end{aligned}\right\} \qquad [d]$$

Indeed for $\sigma < 1$, the bracket value becomes larger than unity, a feature which, because of the presence in the structure of the group $(\sigma - 1)$, is equally shared by Prandtl's Equation [22] and von Kármán's Equation [34] expressions.

The physical implications devolve from comparing Equation [d] with Equation [c]. The bracket groups are obviously equivalent to the reduction coefficient η, which for $\sigma < 1$ becomes thus larger than unity, meaning that a correction term, ostensibly intended to account for the insulating effect of the laminar zone, results in increasing the heat flow numerically beyond the highest limit which could possibly be achieved in accordance with Equation [a] if the whole of Θ_0 were contributing to turbulent convection. Obviously, the source of the discrepancy must lie in the very form of the basic relation of Equation [a], which seems to understate the actual intensity of the heat transfer due to turbulence.

The writer finds, that the inconsistencies are easily removed if instead of using the "momentum," one were to interpret the Reynolds analogy in terms of "energy."[5] In such a case the second of von Kármán's Equations [5] could be appropriately expressed by the ratio

$$C_H = \frac{q}{\rho c U \Theta} = \frac{\text{Energy lost per unit wall surface}}{\text{Energy of flow per unit cross-section area}} \qquad [e]$$

Using for the numerator Equation [10] in a text[6] by the writer, we obtain

$$C_H = \frac{q}{\rho c U\Theta} = \frac{U\tau}{\rho U U^2/2}$$

from which there follows

$$\begin{aligned} q &= 2\,\frac{c\tau\Theta}{U} \qquad [f] \\ C_H &= C_f \end{aligned}$$

By comparison with Equation [a] and Equation [16], the heat transfer derived from interpreting the Reynolds analogy in terms of energy, is twice as large as when using momentum. Physically, there is no reason why the analogy between heat transfer and friction should not be considered in energy terms. In fact convective heat transfer is a process of diffusion, and the

[1] "The Analogy Between Fluid Friction and Heat Transfer," by Th. von Kármán, Trans. A.S.M.E., vol. 61, 1939, pp. 705–710.

[2] Professor, Columbia University. Mem. A.S.M.E.

[3] Numbers in brackets refer to equations in the original von Kármán paper.[1]

[4] "The Mechanical Properties of Fluids," a collective work, Blackie & Son, Ltd., London, 1925, p. 174.

[5] This method of presentation was first disclosed by the writer in his lectures at Columbia University in 1932–1933.

[6] "The Mechanics of Turbulent Flow," by B. A. Bakhmeteff, Princeton University Press, 1936, p. 4.

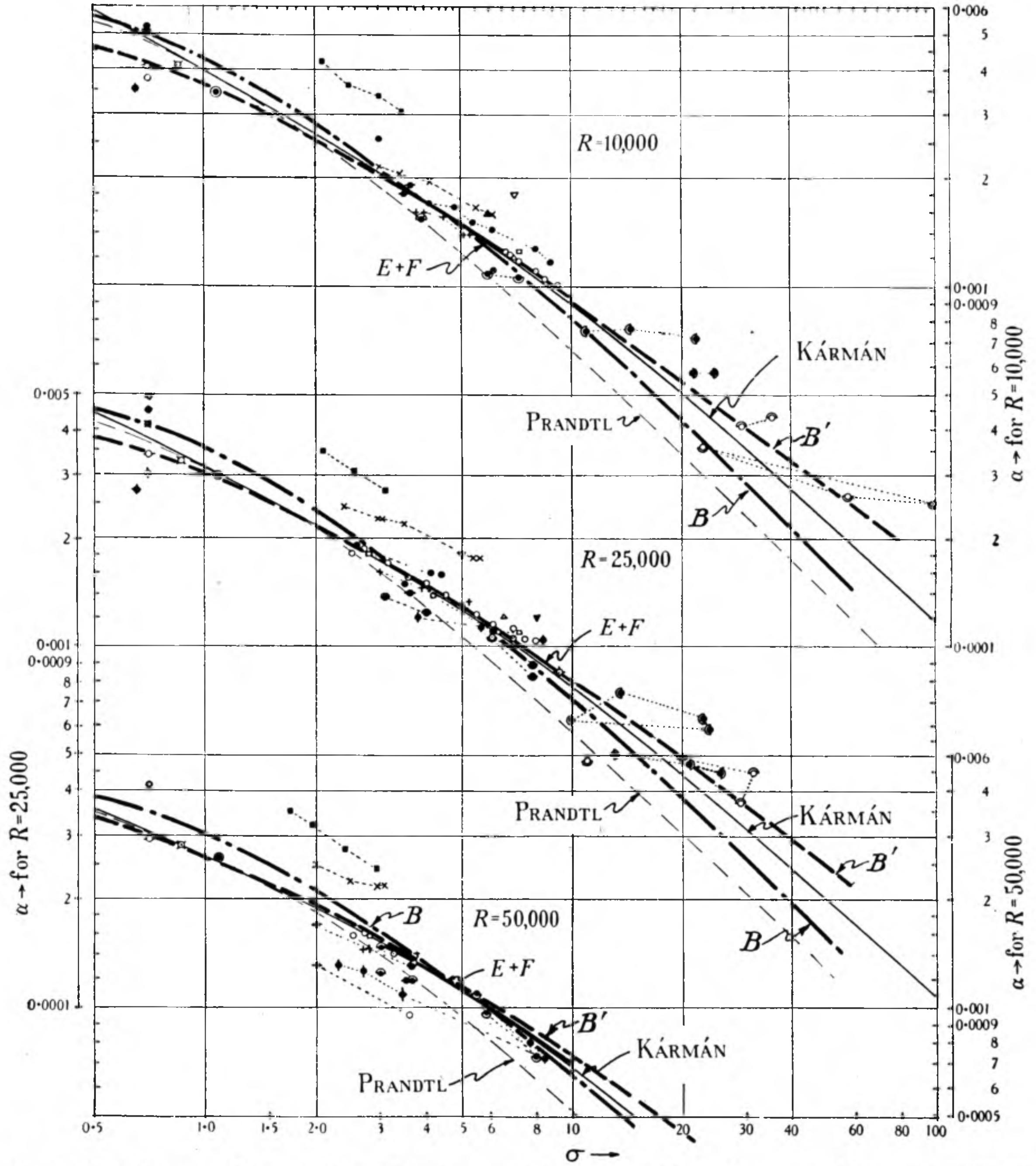


FIG. 1 HEAT-TRANSFER NUMBER IN PIPES, FIG. 4 OF VON KÁRMÁN PAPER,[1] WITH ADDITION OF CURVES REPRESENTING EQUATIONS [h] AND [j]

rate of the latter is appropriately reflected by energy loss.

To arrive at final expression of an equation of the type of Equations [22] and [34] one proceeds by equating the turbulent heat exchange $q = 2c\tau(\Theta\delta - \Theta_{\mathrm{f}t})/U = \eta(2c\tau\Theta_0/U)$ based on Equation [f], with the conduction across the laminar film $q = k(\Theta_w - \Theta\delta)/\delta = (1 - \eta)(\Theta_0 k/\delta)$, in which δ is the thickness of the insulating layer, equivalently expressed through $\delta = \mu U_\delta/\tau$. This determines the reduction coefficient as

$$\eta = \frac{1}{1 + 2\sigma U_\delta/U} \quad \text{[g]}$$

and then with Equations [f] and [c]

$$\left.\begin{aligned} q &= \eta 2\frac{c\tau\Theta_0}{U} = \frac{2c\tau\Theta_0}{U}\left[\frac{1}{1 + 2\sigma U_\delta/U}\right] \\ C_H &= \eta C_f = C_f\left[\frac{1}{1 + 2\sigma U_\delta/U}\right] \end{aligned}\right\} \quad \text{[h]}$$

Assuming finally with von Kármán that, in terms of the dimensionless distance parameter y^* Equation [25], the U_δ/U ratio may be expressed through

$$U_\delta/U = N\sqrt{C_f/2} \quad \text{[i]}$$

with N a numerical constant, one obtains for η the convenient form of

$$\eta = \frac{1}{1 + N\sigma\sqrt{2C_f}} \qquad [j]$$

In first approximation, one single value of N can be assumed to qualify conditions within the whole range of σ and R, the particular number lying ostensibly somewhere near the point, where in Figs. 51 and 53 of Bakhmeteff's text,[6] the u/u_* curves leave the laminar outline. Accordingly in Fig. 1 of this discussion, complementing von Kármán's Fig. 4,[1] the curves, marked B, represent Equations [h] and [j] with N = constant = 7. The practical accord with the experimental points, notwithstanding the crudeness of the assumptions, is quite satisfactory. It does seem, however, that with the increase of σ, the N value, which features the size of the conduction laminar zone, should somewhat decrease. The possible variation could be appraised by a simple exponential expression $N = N_0/\sigma^m$. This possible improvement is introduced into the curves, designated as B', taking N_0 = constant = 9.5, and m = 0.2, so that in Equation [j] $N\sigma = 9.5\,\sigma^{0.8}$.

In all computations, the Blasius values of $\lambda = 4C_f$ for smooth pipes were used.

The insulating effect of the laminar film is especially evidenced by the comparative outlines of the η curves which are plotted in Fig. 2 of this discussion. Obviously with σ approaching zero, the insulating effect vanishes, and C_H tends to its limiting value C_f.

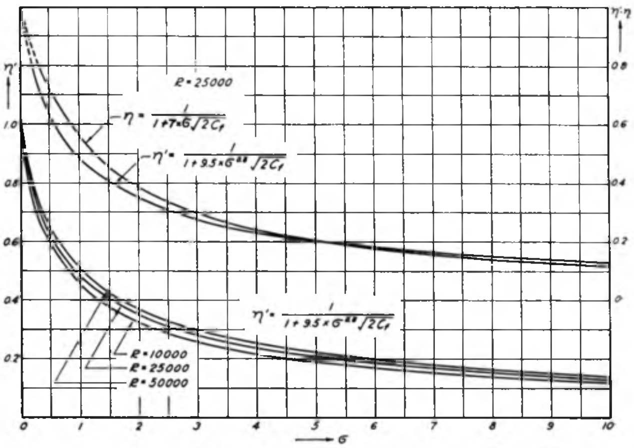


FIG. 2 INSULATING EFFECT OF LAMINAR FLOW EVIDENCED BY COMPARATIVE OUTLINES OF η CURVES

Pump Discharge Valves on the Colorado River Aqueduct

By R. M. PEABODY,[1] LOS ANGELES, CALIF.

This paper outlines the arrangement and physical equipment of the five pumping plants installed on the Colorado River Aqueduct. Studies and estimates indicated that for these aqueduct plants a common delivery pipe with a valve at each pump was more economical than separate delivery lines. Rotating-plug-type pump valves were finally selected for shutoff service at all plants. With proper control of the rate of closure they can be made to limit the pressure rise to a moderate percentage in systems comparable to those of the Metropolitan Water District. Constructed by two manufacturers for application at different plants, the valves, while varying somewhat in detail, give practically the same operating result. Model tests were conducted by the manufacturers and by the hydraulic laboratory at the California Institute of Technology. Details of the design, construction, and operation of the valves, as well as the test results, are included in the paper.

THE Colorado River Aqueduct extends from the Colorado River across the entire width of California and is to supply water for domestic and industrial purposes to the highly developed areas of the south coastal basin of Southern California. The main aqueduct is 242 miles in length. The greater part lies across a barren desert country at a higher elevation than the Colorado River. To cross this elevated region a succession of pumping plants, five in number, are required with a total lift of 1617 ft. The designed capacity of the aqueduct is 1600 cfs. Power for pumping, which for the ultimate development will be about 300,000 kw, is obtained from the power plant at Boulder Dam about 150 miles north of the point of diversion on the Colorado River. Fig. 1 shows the location and the profile of the portion of the aqueduct which includes the pumping plants.

Pumping Plants

Each pumping plant will ultimately contain nine pumping units of 200 sec-ft capacity each, which will allow one spare unit. Three such units are installed in each plant initially and others will be added from time to time as the demand for water increases. Table 1 gives the principal engineering data for each of the plants.

The five pumping plants are almost identical in general design. There are differences in detail made necessary by the variation in head and by the different intake and discharge conditions at the site of each plant. The pumps are all of the single-stage, single-suction, vertical-shaft, volute type, without guide vanes or diffusers. The pumps are set below the minimum inlet water levels, not only to avoid the necessity of priming apparatus, but to give sufficient positive inlet pressure to insure operation at minimum head and maximum discharge without cavitation. The ultimate nine pumps of each plant will be connected, in groups of three, to three main 10-ft diameter, steel delivery pipes.

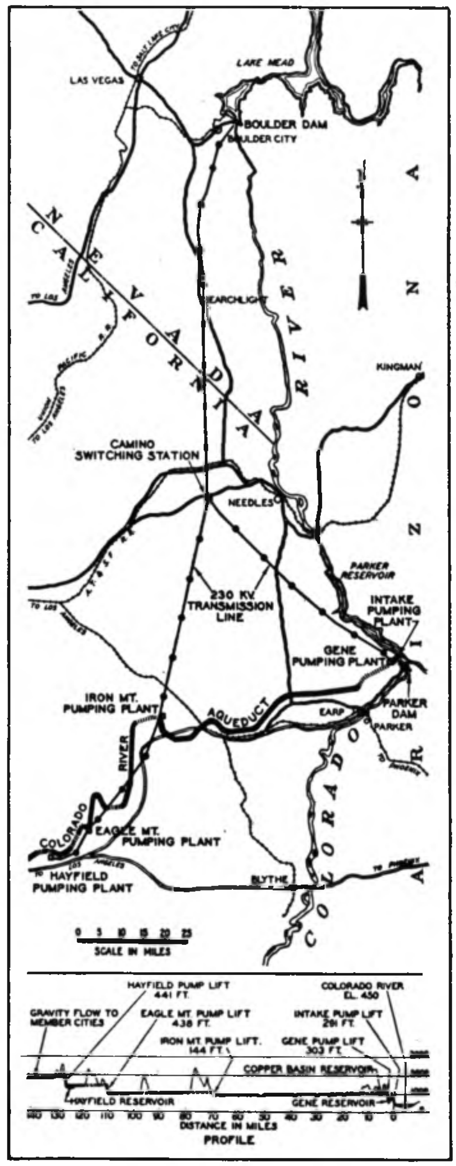


Fig. 1 Map of Power System for Colorado River Aqueduct and Profile of Section Which Includes Pumping Plants

Discharge pipes 6 ft in diameter from each pump converge through a three-branch manifold or wye to the main delivery pipe at a short distance from the plant. The delivery lines at three of the plants, Intake, Gene, and Eagle Mountain, connect at the upper end through surge chambers to pressure conduit or pressure tunnels and reservoirs. At the other two plants, Iron Mountain

[1] Senior Engineer, Metropolitan Water District of Southern California.

Contributed by the Hydraulic Division and presented at the Semi-Annual Meeting, San Francisco, Calif., July 10–15, 1939, of The American Society of Mechanical Engineers.

Discussion of this paper was closed September 1, 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.

TABLE 1 PRINCIPAL ENGINEERING DATA CONCERNING PUMPING PLANTS

Items	Intake	Gene	Iron Mt.	Eagle Mt.	Hayfield
Location - miles west of Intake	0	2	69	110	126
Static Lift	291	303	144	438	441
Average Operating head	294	310	146	440	444
Elev. hydraulic grade at top of lift	740	1,037	1,047	1,404	1,807
Delivery lines - slope length (center line pumps to center line surge tank)	946	2,185	689	947	1,287
Diameter of main lines	10'-0"	10'-0"	10'-0"	10'-0"	10'-0"
Plate thickness (min. to max.)	3/8"-7/8"	3/8"-7/8"	3/8"-5/8"	3/8"-1-1/8"	3/8"-1-1/8"
Diameter of branch lines	6'-0"	6'-0"	6'-0"	6'-0"	6'-0"
Plate thickness (min. to max.)	9/16"-5/8"	5/8" only	1/2"-9/16"	3/4"-7/8"	3/4"-7/8"
Time for round trip of pressure wave, seconds ($^{2L}/_a$)	0.664	1.533	0.485	0.638	0.850
Pumps - Manufacturer	Byron Jackson Co.	Byron Jackson Co.	Allis-Chalmers Mfg.Co.	Worthington Pump & Machinery Corp.	Worthington Pump & Machinery Corp.
Capacity at Rated Head, g.p.m.	90,000	90,000	90,000	90,000	90,000
Speed, (r.p.m.)	400	400	300	450	450
Specific speed (g.p.m. units)	1,690	1,624	2,140	1,403	1,395
Motors - Manufacturer	Gen'l Electric Co.	Gen'l Electric Co.	Allis-Chalmers Mfg. Co.	Westinghouse Elec. & Mfg. Co.	Westinghouse Elec. & Mfg. Co.
Horsepower (each motor)	9,000	9,000	4,300	12,500	12,500
Flywheel Effect, motor and pump (WR^2)	468,000	468,000	339,000	506,000	506,000
Pump Discharge valves, Manufacturer	S. Morgan Smith Co.	S. Morgan Smith Co.	S. Morgan Smith Co.	Pelton Water Wheel Co.	Pelton Water Wheel Co.
Valve size (inlet and outlet dia.)	42" x 58-3/8"	42" x 58-3/8"	48" x 64-1/2"	40-1/2" x 53-7/8"	40-1/2" x 53-7/8"
Inlet Reservoir - useful capacity	Parker Reservoir 387,000 acre-feet	Gene Reservoir 3,000 acre-feet	100 acre-feet	112 acre-feet	Hayfield Reservoir 86,500 acre-feet
Discharge Reservoir - useful capacity	Gene Reservoir 3,000 acre-feet	Copper Basin Reser. 6,000 acre-feet	112 acre-feet at Eagle Mountain	Hayfield Reservoir 86,500 acre-ft.	Cajalco Reservoir 100,000 acre-feet

NOTE: The average submergence of the pump center line below the inlet water level is as follows for the various plants: Intake, 14 ft; Gene, 17 ft; Iron Mt., 15 ft; Eagle Mt., 20 ft; Hayfield, 37 ft.

and Hayfield, the delivery lines connect through simple transition structures to gravity-flow tunnels. A headgate is installed at the upper end of each delivery line so that any main pipe line can be unwatered independently and without draining the reservoir.

It is not considered desirable to sacrifice efficiency by operating the pumps with throttled discharge. The flow in the aqueduct will therefore be in multiples of about 200 sec-ft, the unit pumping capacity. Due to variation in reservoir and canal water levels, the pumps at consecutive plants may operate at heads above or below the design point with a consequent variation in discharge. The reservoirs at Gene draw, Copper basin, Hayfield, and Cajalco are sufficient to hold the entire capacity of the aqueduct between these reservoirs and the next pumping plant upstream. At these plants, variation in pump discharge may be compensated for by starting and stopping one or more pumps at comparatively long intervals, and in power outages of long duration no water will be wasted from the aqueduct. The available regulatory storage at Iron Mountain and at the inlet of Eagle Mountain is small (about 100 acre-feet at each plant) so that starting and stopping of one pump at each of these plants may be necessary at shorter intervals to maintain a balanced discharge. These small reservoirs will provide rejection storage for power outages up to 30 or 45 min duration and, for unanticipated longer outages, spillways provide for wasting the excess flow.

Figs. 2 and 3 show the general arrangement of the Intake plant and appurtenant structures. Fig. 4 shows a cross section of the Intake plant in more detail and is typical of all the plants, except for the arrangement of the pump inlet. In all of the other plants, the inlet is through a branch from a 16-ft-diameter steel manifold. There is a butterfly valve instead of a slide gate to close off the pump inlet.

With the adopted arrangement of three pumps discharging into a common delivery pipe, valves at the outlet of each pump are necessary. Another possible arrangement, which was studied and abandoned because of greater cost, was to have separate delivery lines from each pump all the way to the upper reservoir or conduit with no discharge valves at the pumps, but with a device at the upper end of each pipe line to prevent backflow from the reservoir. With such an arrangement, should a pump be taken out of service, either intentionally or as a result of emergency shutdown, the water would run back through the pump until the delivery pipe was empty. The pump would of course reverse its rotation under these conditions. Studies and estimates showed conclusively that for the aqueduct plants the common delivery pipe with a valve at each pump was more economical than separate delivery lines.

Discharge Valves

The large size of the pumps and the high discharge velocities called for careful consideration of the various available types of valves, which include plain swing-check valves, combinations of check valves, and pressure- or mechanically operated relief valves, butterfly valves, gate valves, needle and plunger valves, and plug-type valves or cocks. On account of the high velocity at the pump discharge, reaching 24 fps at maximum flow, loss of head is an important consideration. Butterfly valves, needle and plunger valves, and certain types of plain gate valves all create a certain amount of obstruction or disturbance in the flow, with more or less head loss. However, two types of valves, the ring-follower gate valve and the rotating-plug-type valve, could be so constructed that at full opening they would offer no more resistance to flow than an equal length of straight or tapered circular pipe. In both types, when fully open, the water passages through the valve body and the valve disk or plug are circular and of uniformly increasing diameter from the pump-discharge flange to the connection of the valve to the delivery pipe.

Of even greater importance than the type of valve is the operating and control mechanism. Considerable power is necessary to operate high-pressure valves of this size, particularly for the rapid valve movement on emergency closure. Electric power is not suitable since failure of power supply would leave the valves inoperative at the very time when emergency operation is necessary. Hydraulic operation, using the delivery-line water pressure, is not wholly desirable, partly because of the variation in the pressure source during closure and even more so because of the possibility of incrustation and corrosion of the small control valve parts and the resulting irregularity or failure in operation. The most reliable power source, although the most expensive, appeared to be hydraulic cylinders, using oil from air-pressure-accumulator tanks similar to those commonly used for supplying power to actuator-type governors for hydraulic turbines.

Specifications were issued for both ring-follower gate valves and rotating-plug-type valves. Hydraulic-cylinder operation was required using oil pressure. The control mechanism was specified to be flexible enough to give the slow valve travel desired for normal opening and closing, as well as an adjustable fast travel for emergency closure. The bids received showed a lower cost for the rotating-plug-type valves in all plants and contracts for this type were awarded to the S. Morgan Smith Company for the valves at the Intake, Gene, and Iron Mountain plants, and to the Pelton Water Wheel Company for the valves at the Eagle Mountain and Hayfield plants. The hydraulic-cylinder operators for the Eagle Mountain and Hayfield valves were made by the Chapman Valve Company for the Pelton Water Wheel Company. Controls for all valves were made for

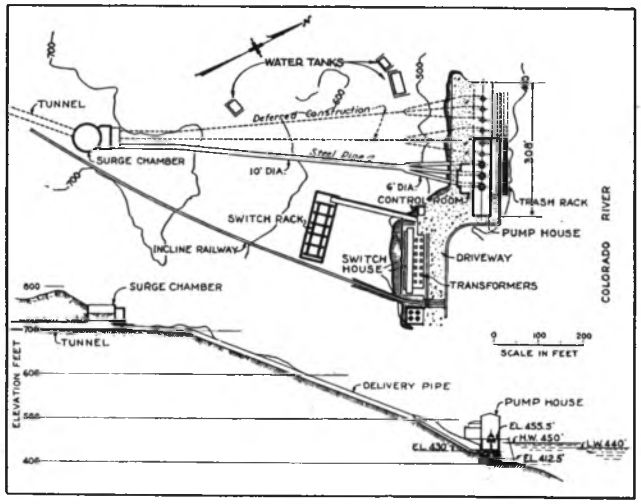


Fig. 2 Plot Plan and Profile of Intake Pumping Plant

Fig. 3 The Intake Pumping Plant on the Colorado River

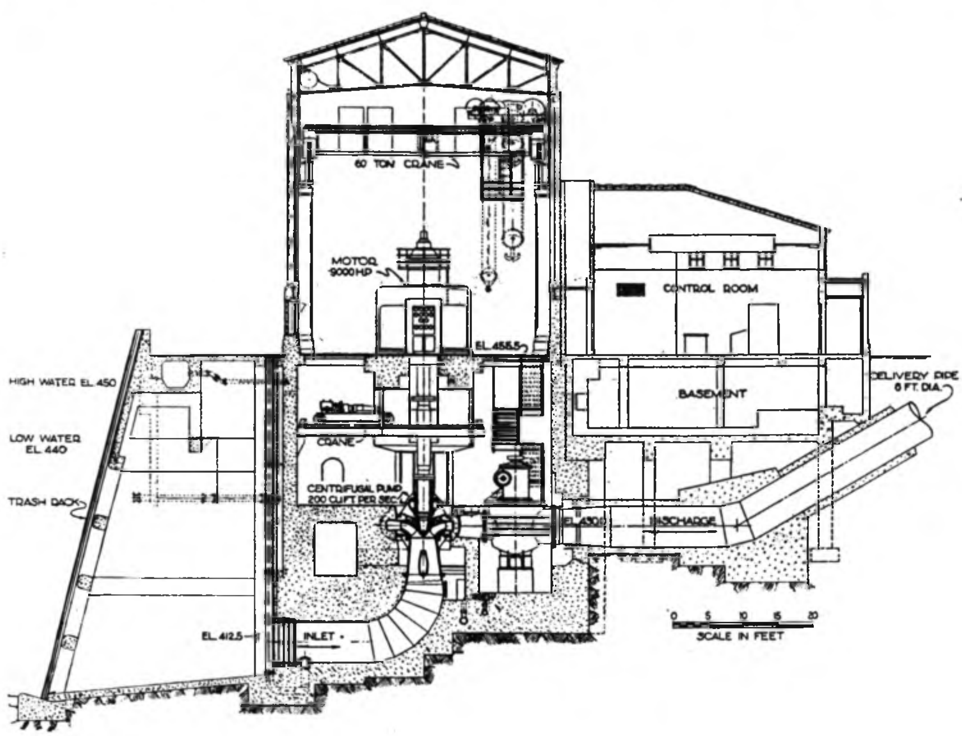


Fig. 4 Transverse Section of Intake Pumping Plant

Fig. 5 Assembling the Body and Plug of a 42-In. by $58^3/_4$-In. Tapered Rotovalve; S. Morgan Smith Company

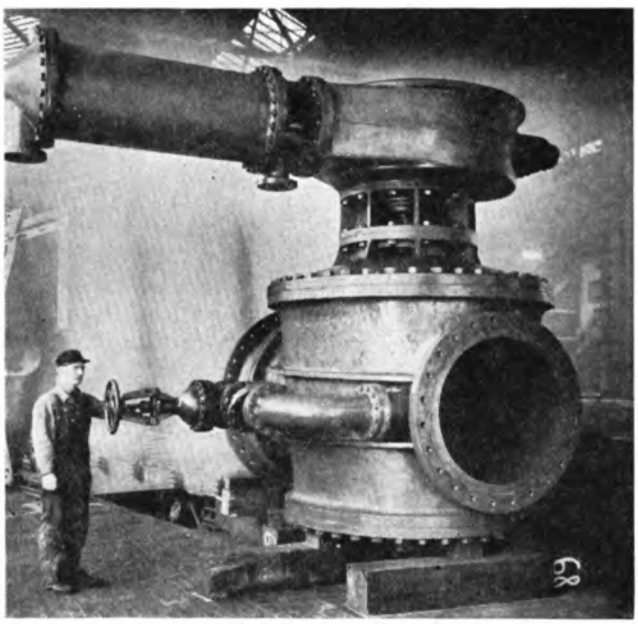


Fig. 6 Complete Assembly of a $40^1/_2$-In. by $53^7/_8$-In. Reseating Hydraulic Cone Valve; Pelton Water Wheel Company

both principal contractors by the Woodward Governor Company.

The general design of the valves made by the two companies is similar, except for the operating mechanism. In all valves, with the plug in the fully open position, the water passage is circular in cross section, increasing in diameter from the pump to the delivery pipe. The total angle of the taper is about 8 deg. The valve thus forms an extension of the pump diffuser and helps to regain the velocity head at the pump discharge. Close to the periphery of the water passage on both the valve plug and valve body are noncorrosive seats, which are of monel metal on the S. Morgan Smith valves and bronze on the Pelton valves. The plugs are conical and, in both the open and closed position, the plug is pushed down into the valve body so that the seats are in contact. To open or close a valve, the entire plug is first raised sufficiently to separate the seats, then rotated 90 deg, and finally pushed down again to reseat. On the sides of the plug are seats corresponding to those around the water passages. There is thus full seat contact in both the open and closed positions. Fig. 5 is a shop erection view of one of the S. Morgan Smith valves which shows the opening and closing seats on the plug. Fig. 6 is an

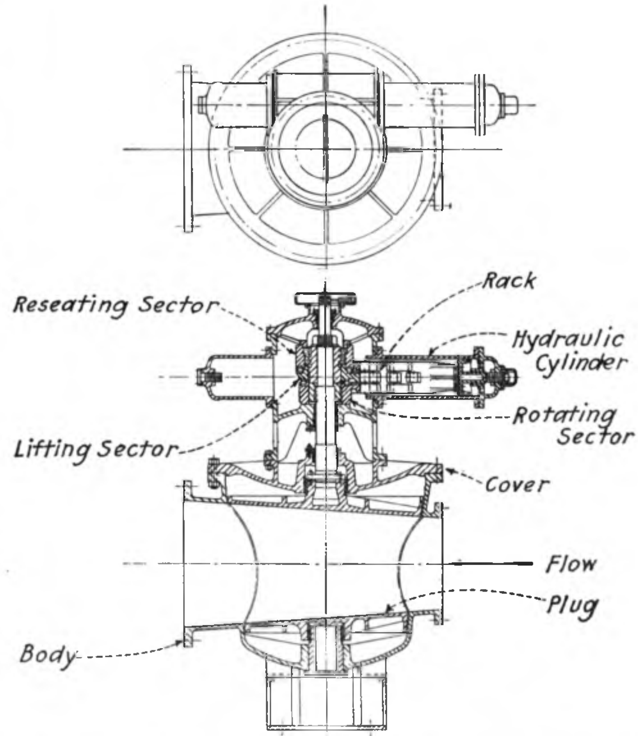


Fig. 7 Cross Section of 42-In. by $58^3/_4$-In. Tapered Rotovalve; S. Morgan Smith Company

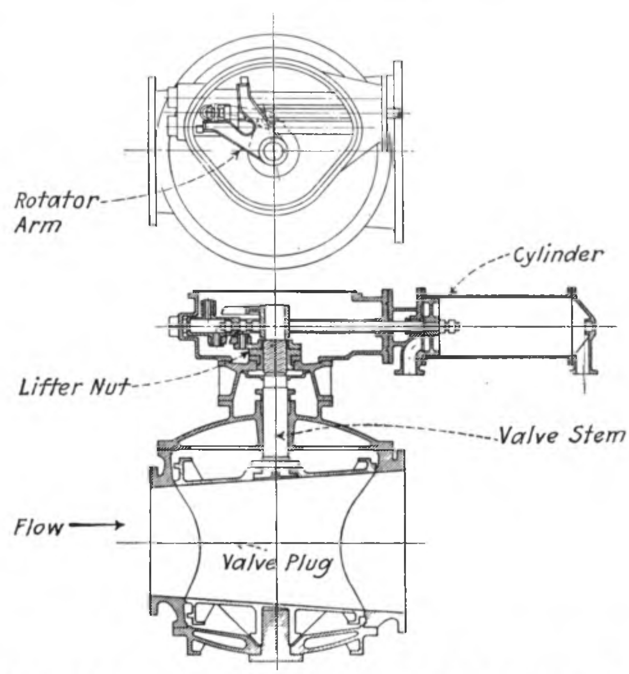


Fig. 8 Cross Section of $40^1/_2$-In. by $53^7/_8$-In. Reseating Hydraulic Cone Valve; Pelton Water Wheel Company

assembled view of one of the Pelton valves. Figs. 7 and 8 show cross sections of the two valves and their operating mechanism.

The valves are all hydraulic-cylinder-operated with oil at 300 lb per sq in. working pressure. The principal difference between the S. Morgan Smith valves and the Pelton valves is in the mechanical arrangement by which the straight-line motion of the hydraulic piston is made to raise, rotate, and then lower the plug.

On the Pelton valves, the operating cylinder is separate from the housing containing the mechanism for rotating the plug. A guided crosshead on the piston rod operates a rotator arm and a lifting nut which are attached to the valve stem. For the first few inches of crosshead travel, the rotator arm is prevented from rotating. Through a linkage the crosshead turns the nut on the threaded portion of the stem in a direction to lift the valve stem and plug. After the crosshead has traveled far enough to lift the plug completely free, the rotator arm is released and a pin on the crosshead enters a slot in the rotator arm. Further travel of the piston rotates the entire assembly of rotator arm, lifting nut, valve stem, and plug. After the required 90-deg rotation, the rotator arm strikes a stop which prevents further angular motion; the pin on the crosshead comes out of the slot in the rotator arm; and further travel of the piston turns the nut in a direction to lower the valve stem and reseat the plug. The crosshead, rotator arm, and lifting nut operate in a bath of lubricant.

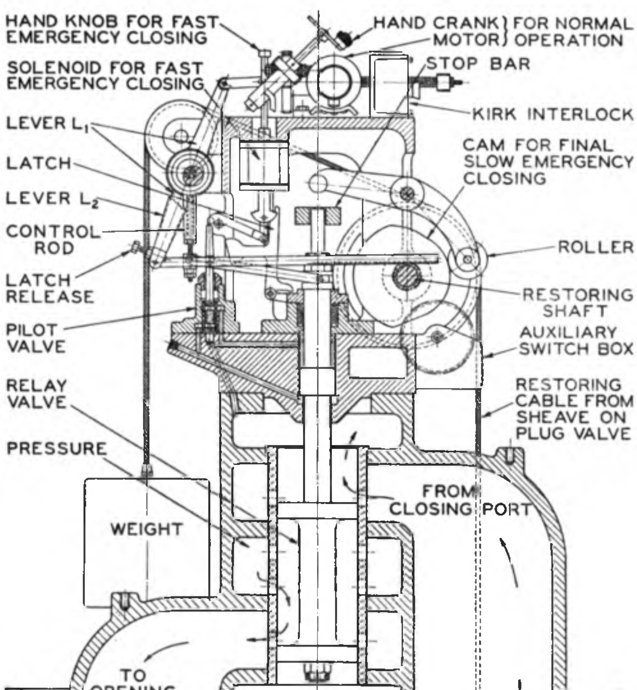


Fig. 9 Cross Section of Control Mechanism Used on All Valves; Woodward Governor Company

On the S. Morgan Smith valve, the lifting, rotating, and reseating motions are accomplished through an interacting combination of racks and gear segments. The piston acts directly on a rack which has three separate rows of teeth, each row matching with a separate gear sector. The first few inches of piston travel cause rotation of the lifting gear sector. The lifting sector is keyed to a screw which rotates freely on the valve stem but is threaded into a nut which is keyed to the reseating gear sector. During this portion of the stroke, the rotating sector and the reseating sector are both prevented from turning by special teeth which slide on flat surfaces on the rack. When the stem has lifted the required amount, all three gear sectors are engaged by the rack teeth and rotated 90 deg. The rotating sector and the lifting sector then come out of mesh with the rack and are prevented from rotating further by stops in the housing. The reseating sector is still in mesh, however, and the relative motion between the nut and the screw forces the nut downward and reseats the plug. Adjusting screws limit the rack travel for both seating and reseating and can be adjusted to give the proper seat contact.

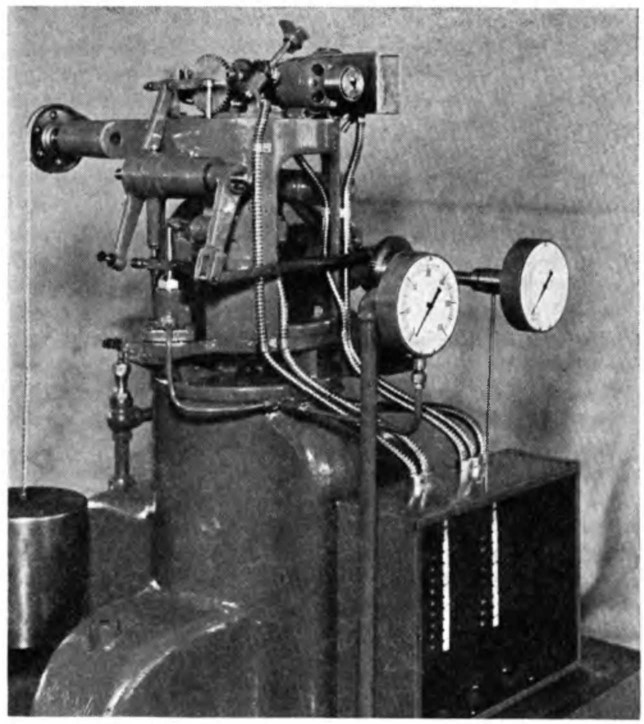


Fig. 10 Assembly of Control Valve, Cover Removed; Woodward Governor Company

The principal parts of all the valves are steel castings fully annealed. Shop hydrostatic tests were made on all valves at a pressure approximately two and one-half times the maximum static head. The operating cylinders were tested at 450 lb per sq in. oil pressure. The specifications required the operating mechanism to have sufficient capacity to unseat the valve against an unbalanced pressure of 120 per cent of the maximum static head, assuming a friction coefficient of 0.50 for the seats, and a pressure of 250 lb per sq in. in the operating cylinder.

Control Mechanism

The source of oil pressure for valve operation is an air-pressure-accumulator tank, with return-oil sump tank and pump, very similar to the oil-pressure sets used for actuator-type governors for hydraulic turbines. The proper oil level and pressure are maintained in these tanks automatically. There is an independent pressure set for each valve. The control mechanism, furnished by the Woodward Governor Company for all valves, operates the valves slowly for normal starting and shutting down a pump. On failure of power to a pump motor, this mechanism closes the valve very rapidly to about 5 per cent of the effective opening and then very slowly to full closure. The time of the fast part of the stroke can be varied from about $1^1/_2$ sec to 5 or 6 sec. The point on the stroke where the motion changes from fast to slow can be varied. The slow-closing part of the stroke can be adjusted to any time up to about 60 sec. For normal opening and closing, the time of the full stroke is 40 to 60 sec and the rate is nearly uniform.

Before starting a pump, the inlet valve is fully opened and, since all the pumps are set well below the inlet-water level, the pump casing is completely filled. The discharge valve is closed and normally the discharge pipe above the valve is completely filled. The motor is started, brought up to speed, and synchronized, the pump operating at practically shutoff head. An 8-in. by-pass with a check valve on each of the main plug valves permits sufficient water to be pumped to prevent overheating the water in the pump casing. The by-pass is also used to fill the delivery line if it is empty when the pump is started. The main plug valve is then opened slowly. When taking a pump out of service, the discharge valve is first slowly closed, the motor circuit breaker opened, and the unit decelerates to rest.

Fig. 9 shows a cross-sectional view of the control mechanism. Fig. 10 shows the control valve with the cabinet removed. The relay valve for applying oil pressure to operate the hydraulic cylinder, the pilot valve, the floating lever (in this case a double eccentric), and the control rod are similar in construction and operation to the corresponding parts in a water-wheel governor. The control rod corresponds to the speed rod of the flyballs.

Referring to Fig. 9, the control rod is actuated by a double eccentric. The control rod is journaled on the hub of the outer eccentric and the outer eccentric is journaled on the inner eccentric. The outer eccentric is integral with the lever L_1 and is turned thereby. The lever is operated by the back-geared motor, through gearing and a screw. The inner eccentric is integral with the central shaft, which is turned by the restoring lever, L_2. To the end of lever L_2 is attached a rack bar which meshes with a gear on the restoring shaft. This shaft is connected to the vertical shaft of the main pump discharge valve by a cable and sheaves, the cable and sheaves all being enclosed. Thus lever L_2 moves with and its position corresponds to the angular position of the cone valve plug. The cable is kept taut by a heavy weight. With the levers L_1 and L_2 in line with each other, the control-rod length is adjusted to hold the relay valve in its central position with the relay ports closed. If lever L_1 is moved out of line with L_2, the control rod will be moved up or down, causing the relay valve to be moved down or up and admit oil to the main-valve operating cylinder. The main valve plug then rotates and, through the cable connection, the restoring shaft, and the rack, brings lever L_2 in line again with L_1, moving the pilot valve in a direction to cause the relay valve plunger to close the relay ports and stop the movement of the plug. If L_1 continues to move, then the main valve plug and lever L_2 will continue to move at the same rate.

In Figs. 9 and 10, the levers are shown in the position they assume after the main valve has been fully opened. To close the valve in a normal slow-closing operation, the control motor is started in the direction to cause the upper end of lever L_1 to move to the left, which will cause the main valve plug to unseat and rotate in the closing direction, and lever L_2 to move toward a position in line with L_1. When lever L_1 has been moved to its extreme left position, the main valve plug will have rotated to its fully closed position. A little overtravel of the lever L_1, causes the piston of the hydraulic cylinder to continue on and seat the plug. The control-valve motor is provided with a friction governor to secure a constant speed irrespective of the voltage. The governor is adjusted to give full plug rotation in about 40 sec.

To open the valve the motor is operated in the opposite direction.

In case of motor or current failure, the valve may be operated by the small hand crank shown close to the motor. This is geared for a number of turns calculated to make it convenient to operate at about the same speed as motor operation gives.

Limit switches operated by the motor screw open the motor circuit when the screw has traveled its full movement in either direction.

For emergency closing, the main-valve plug is closed rapidly by energizing the 125-volt, d-c solenoid or by raising the solenoid core by means of the hand knob. As the core is raised, it acts, through the link and lever shown in Fig. 9, to depress the pilot valve. A spring in the control rod permits it to do so. The pilot valve then causes the relay valve to rise, closing the plug rapidly, at a rate determined by the setting of the stop bar, which limits the amount the relay valve can open.

When the solenoid core is raised, the latch shown falls into place and holds it up. Then the plug cannot be moved to open until the latch is released. This will occur when the lever L_1 has been moved to the closed plug position and the lower extension of lever L_1 strikes the latch.

When the plug has been closed rapidly to a predetermined position, the cam, shown on the restoring shaft, engages a roller on the right-hand end of a rocker arm shown just above the cam. The other end of the rocker arm is thereupon depressed and forces the relay valve down to a port opening which will cause the final closing of the plug to be made at the desired slow rate.

The principal adjustments on the control mechanism are as follows: (1) The rate of normal opening or closing can be varied by the governor on the d-c motor; (2) the initial rapid-closing rate can be varied by raising or lowering the stop bar, which limits the travel of the relay valve; (3) the position of the main-valve plug, at which the emergency-closing rate changes from fast to slow, can be varied by rotating the cam with reference to the restoring shaft; (4) the final slow-closing rate can be varied by replacing the roller which is lifted by the cam and using another roller of slightly larger or smaller diameter. Once made these adjustments are locked and are not readily changed.

The entire control mechanism is enclosed in a dustproof cabinet. The top cover of the cabinet is hinged and provided with a lock. When this cover is raised it gives access to the hand control for normal and emergency operation. For access to the adjustments, there are two side openings with dustproof covers, bolted in place from the inside.

Operation

Control of the main-pump motors and of the pump discharge valves is centralized in a control room adjacent to the pumping plant. Normally, the sequence of operations in starting and stopping a pumping unit is automatic. In starting up, the operator throws a switch on the control board which applies full voltage to the main motor. As the speed approaches synchronism, a relay operates to apply field current and the motor pulls into step. Closing the field breaker initiates the opening of the pump discharge valve. For normally shutting down a unit, the operator turns the control switch in the closing direction, which starts the pump discharge valve closing slowly. When the valve plug has rotated past the cutoff position so that the pump discharge is entirely through the 8-in. by-pass, the motor is tripped off by a switch actuated from a cam on the control-mechanism restoring shaft.

On emergency closure, the motor is tripped off and rapid closure of the valve is started at the same time. There is an emergency shutdown switch on the control board in the operator's room, so that the operator can trip the unit off in this manner whenever necessary. Emergency trip of the valve and motor is also initiated by any one of the following abnormal conditions: (1) Failure of lubricating-oil supply to pump or motor bearings;

(2) failure of water supply to the motor coolers; (3) operation of motor overload or differential relays; (4) operation of transformer differential relays; (5) accidental opening of motor or high-voltage circuit breakers; (6) phase or line-to-ground faults on transmission system.

Laboratory Tests

Extensive tests on small gate valves and plug valves and on scale models of the pump discharge valves were made by both of the firms that manufactured the valves and by the Metropolitan Water District. These tests were made in order to estimate closely the power required to operate the valves under the assumed maximum-flow conditions, and to determine the loss in head through the valves at various openings. Tests by the manufacturers were made in their own laboratories. The District's tests were made in the hydraulic-testing laboratory at the California Institute of Technology.[2]

It is not within the scope of this paper to describe in detail the work done in the various laboratories. Preliminary testing by the District was conducted on a 6-in., quarter-turn, S. Morgan Smith plug cock with a circular water passage of uniform diameter and on a 6-in., ring-follower, Yuba gate valve. Tests were confined to determining the loss of head at various openings and rates of flow. Tests by the valve manufacturers were made on scale models of the actual valves as they were to be constructed. The model valves were about one seventh the linear dimensions of the full-sized prototypes. The manufacturers' tests were primarily for determining the torque required to rotate the valve plugs at various openings and rates of flow. However, since the laboratory facilities also permitted head-loss measurements, these were taken for the same openings and rates of flow as the torque. The independent tests made by the S. Morgan Smith Company and the Pelton Water Wheel Company were in reasonably close agreement both as to torque and head loss.

The torque tests showed conclusively that, with the operating mechanism designed to have power enough to unseat the plug under the maximum unbalanced head required by the specifications, there was more than ample torque to rotate the plug and control its rate of rotation at any angular position and under the severest flow conditions reasonably expected, even considering the possibility of free reverse discharge through a ruptured pump casing.

The model valve, made by the Pelton Water Wheel Company and used by that company in the determination of torque and head loss, was afterward made available to the District. Further head-loss tests were carried out on this model in the District's laboratory at the California Institute of Technology. Prior to these tests, seat rings were added to the model in order to make it conform as nearly as possible in dimensional relations with its prototype. However, in the unseated position after lifting the plug and in that portion of plug rotation between cutoff and the full quarter turn, it was not practicable to make the clearances such that the clearance dimensions were in the same ratio to the prototype as the main water-passage dimensions. Consequently, the water that flows past and around the model valve plug, when it is lifted and rotated to the position where the water passage through the plug no longer opens directly into the water passage through the valve body, is not a correct measure of the flow through the corresponding clearances in the full-size valve. Since this so-called leakage could not be segregated and accurately measured and, even if it could be measured, would not represent the correct leakage in the prototype, it was decided to neglect it in water-hammer calculations and to assume that the flow through the valve is entirely shut off when the plug is rotated to the cutoff position. This assumption is on the side of safety, since abruptly cutting off the final flow results in a higher calculated pressure rise.

For both the Pelton and S. Morgan Smith valves, the cutoff position is about 70 deg from the full-open position. During the remainder of the 90 deg of total rotation, the plug is lifted off its seat and there is a considerable flow around the plug and through the clearance between the plug and body seats until the reseating operation has brought the seats into full contact. The effect of neglecting this flow is to make the calculated pressure rise considerably greater than the actual observed pressure rise.

In the first loss-of-head tests made by the District, as well as in similar tests made by the manufacturers, the model valve was placed in a straight run of pipe and the net head loss due to the valve only was measured. Pressures, measured directly downstream of the valve at partial openings, showed a considerable drop, due to the high velocity. Therefore, the net head loss was measured from a point just upstream of the valve to a point sufficiently far downstream (about six pipe diameters) so that recovery of velocity head was no longer observed. In the prototype installation, however, the valve is placed directly at the pump discharge. In reverse flow, the jet from the partially opened valve enters the pump casing before the jet velocity has been fully converted into pressure head. A setup was made in the laboratory to simulate the actual installation. The model valve was placed directly at the discharge of a model pump, the complete head and torque characteristics of which had been determined previously for a wide range of normal and reverse rotation, and direct and reverse flow. From this series of experiments, it was found that the head loss in the valve, as affecting reverse flow through the pump at both normal and reverse rotation, was somewhat less than the head loss when the valve was tested in a straight run of pipe.

The experiments showed that the head loss through the valve can be expressed with sufficient accuracy by the equation

$$H_v = KV^2/2g \qquad [1]$$

where H_v is the head loss in the valve, in feet; V is the velocity through the valve plug at the center of the plug in feet per second; and K is a coefficient which is a function of the valve opening and is determined experimentally.

Fig. 11 shows the values of K, determined by these experiments for the Intake and Gene valves and plotted as a function of the angle of the plug in degrees from the full-open position. It should be noted that K approaches infinity as the plug is turned to the cutoff position.

Control of Water Hammer

In order to meet the construction program, it was necessary to design and construct the pump-delivery pipe lines before all of the valve investigations had been completed. Preliminary water-hammer studies were based upon the first experiments on 6-in., uniform-diameter plug and gate valves and upon pump characteristics taken from a pump, having a specific speed somewhat near that of the full-sized pumps, but not in any sense a scale model. These studies showed that the most severe combination of abnormal operations reasonably to be expected was for the motor to be tripped off, the valve not starting to close, until full reverse flow had developed and then closing for its full stroke at a very rapid rate without any retardation near the end of the stroke. It was assumed that this condition would occur on only one pump in any group of three, the valves on the other two pumps operating normally. The resulting maximum pressure rise was calculated to be about 50 per cent for the two highest-head plants, Eagle Mountain and Hayfield; about 75 per cent for

[2] "The Hydraulic Machinery Laboratory at the California Institute of Technology," by R. T. Knapp, Trans. A.S.M.E., vol. 58, Nov., 1936, p. 663.

the Intake and Gene plants; and about 125 per cent for the lowest-head plant, Iron Mountain. The pipe lines were designed for the maximum total heads given by these overpressures, using a maximum stress of 20,000 lb per sq in. in the 10-ft-diameter pipe and 15,000 lb per sq in. in the 6-ft pipes leading from the pumps and in the manifold connecting the 6-ft pipes to the 10-ft pipe.

Water hammer in the pump-delivery lines is a transient condition occurring within a very short period of time. It results from rapid variations of the velocity. The slower the changes in velocity, the less the pressure variation. The characteristics of the pipe line, the head loss through the valve as it closes, the variation of pump discharge and torque with changing head and speed, and the combined flywheel effect of the pump and motor, all affect the velocity in the pipe line and hence the magnitude of the pressure variation. The water-hammer pressure produced by an emergency shutdown of a pump is thus the result of interaction between the pump and motor, the pipe line, and the shutoff valve. Analysis to predict water-hammer pressure must take into account the characteristics of all these elements of the pumping system.[3] The motor, pump, and pipe-line characteristics, once selected, are fixed, so that the timing of the valve stroke is the only available means of regulating the velocity changes in the pipe.

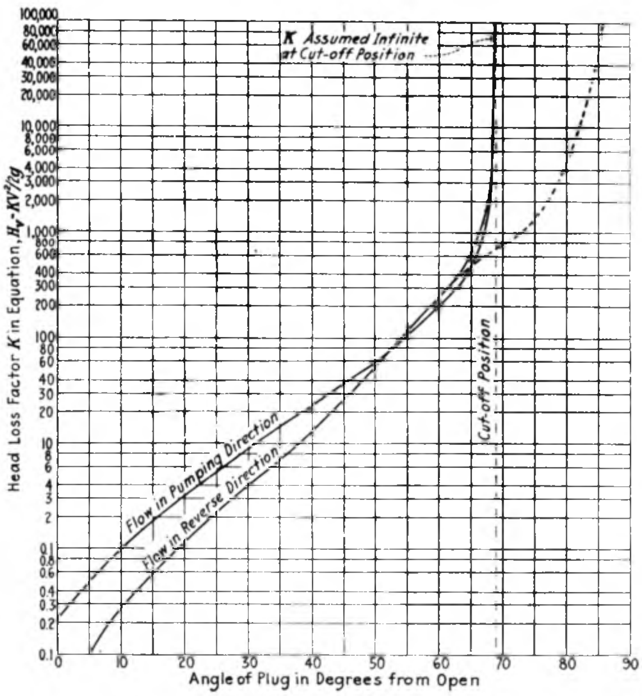


Fig. 11 Head-Loss Coefficient K, Plotted as a Function of Valve Opening

The valve timing for emergency closure is adjusted so that closure is very rapid for about 59 deg of plug rotation, reducing the effective valve area to about 5 per cent in about 3 sec. During this period, the pump is rapidly decelerating, the velocity in the pipe line drops, and the pipe-line pressure drops. As shown by the K diagram in Fig. 11, the throttling effect for the first part of the stroke is small. The pressure drop during this period is dependent almost entirely on the pump characteristics, the motor flywheel effect, the length of the pipe line, and the initial velocity. The valve has comparatively little influence since, by the time the valve closes to an appreciable throttling effect, the velocity is low whether forward or reverse. Somewhat before the time the valve reaches the end of the rapid part of its stroke, the pressure beyond the valve has begun to build up due to reflection from the upper end of the pipe line. The speed of the pump in the meantime has dropped to a point where the head developed by the speed of rotation is less than the opposing pressure. The flow reverses, although the pump is still rotating in the pumping direction. The reverse flow through the pump acts as a relief for the overpressure which tends to develop in the pipe line. The very slow final closure of the valve prevents any large pressure rise in shutting off the reverse flow. By adjusting the final slow-closing time, it is found possible to make complete closure before reverse rotation of the pump, and yet without producing overpressure of any considerable magnitude. Fig. 12 shows the valve timing used at the Intake and Gene plants.

Analyses were made for a number of different emergency conditions, such as delayed tripping of the valve until full reverse flow is developed, abnormal initial speed drop of the motor due to electrical trouble, emergency valve closure without tripping off the motor, and other possible abnormal operations. In all cases the calculated maximum pressure rise was well below the maximum

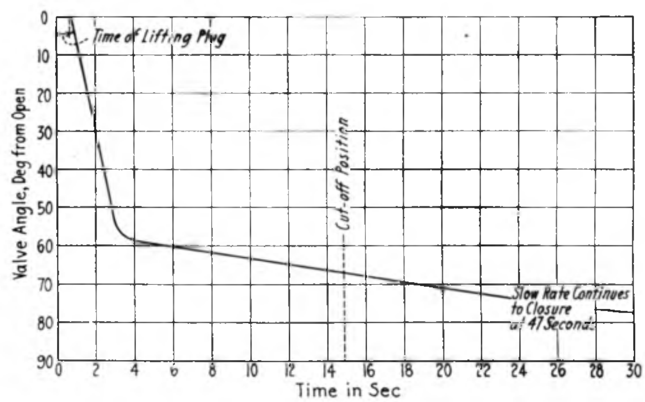


Fig. 12 Time Rate of Change of Valve Opening for Emergency Closure at Gene Plant

(Timing shown is very close to the actual valve timing recorded simultaneously with the test curves in Figs. 13 and 14.)

which the pipe lines and other apparatus are designed to withstand, although it was found that in some cases the pump would reverse rotation and attain considerable speed in the reverse direction.

Comparison of Calculations With Field Tests

At the date of writing (February, 1939), the first two pumping plants, Intake and Gene, had been placed in operation, and a number of observations had been made on the water hammer produced by several different types of emergency shutdowns. Comparison of the observed pressure rise with previous calculations shows that in every case the observed pressure rise is less than that calculated. Figs. 13 and 14 show the comparison for one and three pumps tripped off at the Gene plant, where there is the longest delivery line. One reason for the difference between the observed and calculated results is undoubtedly the assumption made in the calculation that the flow is abruptly cut off when the plug is rotated to cutoff position. Reference to Fig. 12 shows the K curve approaching infinity at cutoff. Probably the K curve in that region is more nearly that shown by the dotted line, indicating that the flow just before cutoff is more than was assumed and that flow continues and is gradually shut off between cutoff position and full rotation.

[3] "Typical Analysis of Water Hammer in a Pumping Plant of the Colorado River Aqueduct," by R. M. Peabody, Trans. A.S.M.E., vol 61, Feb., 1939, p. 117.

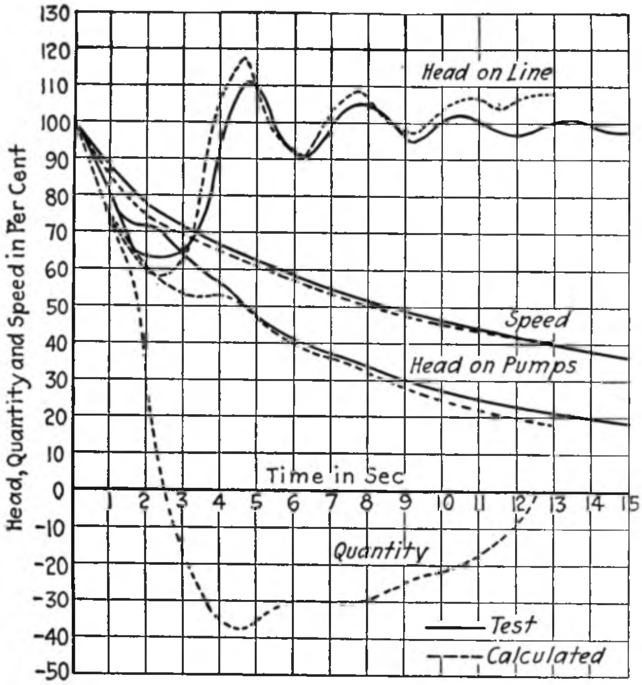


FIG. 13 COMPARISON OF MEASURED PRESSURE AND SPEED VARIATION WITH CALCULATIONS AT THE GENE PLANT; EMERGENCY SHUTDOWN OF ONE PUMP; NO OTHER PUMPS OPERATING

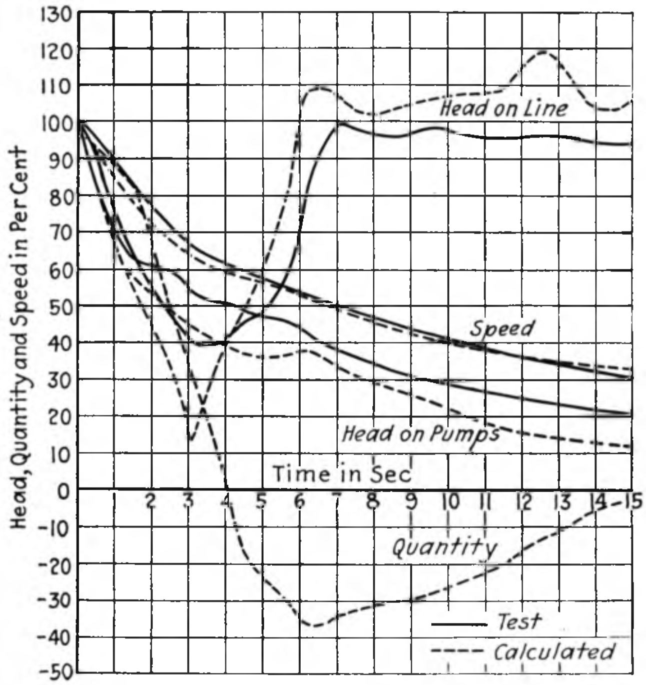


FIG. 14 COMPARISON OF MEASURED PRESSURE AND SPEED VARIATION WITH CALCULATIONS AT THE GENE PLANT; EMERGENCY SHUTDOWN OF THREE PUMPS SIMULTANEOUSLY

It is known from observation that considerable reverse flow continues after cutoff but it has not been possible to measure it. The closing time of the three valves was purposely made slightly different so that all the valves do not fully close at the same instant. This tends to reduce further the pressure rise when two or three pumps are tripped off simultaneously. A careful study of the speed drop at the very start of the transient, which the scale of Figs. 13 and 14 is too small to bring out, shows that, in the first small interval after the start of the transient, the drop in speed is less than given by calculation. This would tend to reduce the initial pressure drop and consequently the subsequent pressure rise. After the first half second or so, the observed and calculated speed drops check fairly well. This indicates a larger flywheel effect at the very start. A suggested explanation is that the rotational inertia of the water in the casing surrounding the impeller tends to increase the total inertia of the pump and motor, and that this effect diminishes rapidly with reduced discharge.

In Fig. 13 the single pump shutdown was made from the emergency switch in the control room. The three-pump shutdown was made by opening the 230-kv circuit breaker close to the plant. The most severe pressure rise observed was when a single pumping unit, no other units operating, was tripped by opening the circuit breaker at the Boulder power plant, with 237 miles of energized 230-kv transmission line between that breaker and the pumping unit. In this case the motor had to act momentarily as a heavily loaded generator supplying charging current to the line. The effect was a rapid drop in the speed of the motor in the short interval before the reverse power relays could act to open the motor circuit breaker and clear the motor from the line. With several motors operating in parallel, each would contribute only a part of the charging current and the speed drop would be less rapid.

Line-to-ground and phase-to-phase short circuits were applied to the transmission system at different distances from the plant, but the resulting pressure rise was less than in the case above cited.

When one pump was tripped off, while either one or both of the other pumps in the plant continued to run, the pressure rise was found to be less than when the pump was tripped while the other two pumps were shut down. This was to be expected since with several pumps running the return-pressure wave, resulting from shutting down a single pump, is relieved through the operating pumps.

Pressure variation in the pumps and pipe line was measured by a fast-motion, strip-chart pressure recorder. The records of valve movement were taken on a fast-motion recorder. Speed variations were observed on a high-speed recording voltmeter, which showed the variation in the voltage of a magneto-tachometer driven from the motor shaft.

CONCLUSION

Rotating-plug-type valves are well adapted for shutoff valves on large pumps. With proper control of the rate of closure they can be made to limit the pressure rise to a moderate percentage in systems comparable to those of the Metropolitan Water District. A marked advantage of this type of valve, which is lifted off its seat while rotating, is that the entire flow is not abruptly shut off when the main water passage is rotated past the cutoff point. The final flow is thus extinguished gradually which tends to cushion the maximum pressure rise.

The magnitude of the pressure rise to be expected can be predicted by well-established methods of calculation, provided that the characteristics of the elements making up the pumping system are known.

All work on the Colorado River Aqueduct is under the direction of F. E. Weymouth, general manager and chief engineer of The Metropolitan Water District of Southern California. The design and construction of the pumping plants are under the direction of J. M. Gaylord, chief electrical engineer.

Discussion

Robert W. Angus.[4] The writer wishes to commend the author for making available some of the data and results used in connection with the water-hammer study on this very extensive pumping problem. The magnitude of the project is such as to justify experimental investigations of the characteristics of the pumps and also of the valves used, and information is thus available in a somewhat unknown field.

The paper is really part of the one by the same author,[3] where the method of working out the water-hammer pressures is given in some detail. For a proper study of this nature, a complete set of characteristic curves of the pump is necessary, covering the action of the machine (1) as a pump, (2) running forward but with the water running downward through it, (3) running under zero torque with the water passing downward, and (4) operating as a turbine with both water and impeller running in reversed directions. The excellent work[2] of R. T. Knapp, to which the author refers, has given data on a pump in such a complete form that its action on water hammer may be studied. It is hoped that similar information will be made available on a number of other pumps.

Presumably, the valve referred to in the author's earlier paper[3] is similar to that in the present one; in any event, the characteristics of the two are of the same form, and the data in the papers supplement one another. In Fig. 11 of the present paper, the losses in the valve for different plug positions are shown, the loss naturally varying with the direction of flow. The head-loss coefficient K will vary with the reference velocity, which has here been taken as that at the center of the plug, whereas it would seem to be more consistent to use one of the end velocities. For partial openings the plug will not be filled with water and the meaning of the velocity there is obscure; the author apparently means by it the quotient of the discharge divided by the area of circular opening at the center of the plug.

The writer has made a number of experiments[5] on a 3-in. cone valve with a cylindrical opening and, while it is not possible to compare the coefficients on this small valve with those on such a large tapering valve, yet there is in general a very good agreement, particularly for the large openings. In the small valve the cone was not lifted off its seat in any position, and hence the smaller openings would be less in agreement.

Fig. 12 shows that in the Gene plant, for the pipe line of which the value of $2L/a$ is given as 1.533 sec in Table 1, the plug has been turned through about 58 deg in about 3 sec or, if allowance is made for the time of lifting the plug, the latter has been turned through about 58/69 or 85.5 per cent of its total closing movement in about 2.3 sec or 1.5 times $2L/a$ sec, which is very quick. The remaining 14.5 per cent of the movement has been made in about 12 sec and Fig. 14 shows that this causes no pressure trouble when the water is running at full velocity in the delivery line. This further illustrates the point mentioned by the writer in the paper[2] already referred to. Many statements have been made that valves in long pipe lines must be closed very slowly to avoid water hammer, whereas the author shows that a closure in 15 sec causes no appreciable pressure rise.

Provided that it is not less than perhaps 10 times the period of the pipe, the total time of closure of a valve is of far less importance than the way in which the closure is effected. For many types of valves it is scarcely possible to effect the first 70 per cent or so of the closure fast enough to cause any trouble, but the final part of the operation must usually be done with great care and with proper consideration of the characteristics of the valve being used. Great differences exist among valves, in regard to their method of control, and some would cause trouble under similar operating conditions to those herein mentioned. However, the valve described in the paper has good features, and the author has done a real service to the profession in giving information about it in such detail.

The writer feels that the control valve shown in Figs. 9 and 10 would be very expensive and quite impossible for any but the very large plants. There may, however, be ways of effecting a similar movement of the cone in a simpler way and, if so, the writer would like information about them. The writer also thinks that some details of the recording devices used in producing the curves of Figs. 12, 13, and 14 would be of much help to those working on water-hammer problems, and hopes the author will add this information to his paper.

A. Hollander.[6] This paper is a worthy complement to the author's former paper,[3] disclosing the method of analysis and the means of reducing the pressure rise in a large pumping system. It shows that a relatively inexpensive rotating-cone valve, which continues the diffusion of the pump-discharge taper without any additional loss under normal operating conditions, is the only direct regulating means required for such a system. The indirect means are a constant-pressure oil supply and the Woodward actuator-type governor. This is the first application of this turbine governor in a pumping plant, where it serves as a pilot valve for the oil-pressure-operated cone valve. The very complete model experiments included the combined pump-and-valve behavior under transient conditions. In reference to these, the writer would like to ask whether both directions of rotation from open to closed were explored, as they would undoubtedly give different results for reverse flow, due to the different entrances into the volute.

It is interesting to note that the valve coefficient K (should be designated as φ)[7] is increasing from open (0 deg) to closed (68 deg) with an inflection point at around 55 deg. If this coefficient is referred to the gate area actually open, $H_v = \dfrac{A \text{ gate}}{A} c \dfrac{v^2}{2g}$ (it should be corrected to read this way in the first paper),[3] then the coefficient c shows a maximum at about 38 deg from open.[8] The two coefficients with the areas at different openings and time rate of opening give a complete picture of the valve effect.

The writer would suggest that in the time-history diagrams[9] the term "quantity" be changed to "flow" or "flow rate" as this is what it means. Furthermore, it should be understood, that this flow, as a function of time, refers only to the flow at the point where the pressure is measured, i.e., to the sections next to the valve, "head on line" at the pipe-line side, and "head on pump" at the pump side of it. For these sections, the flow being proportional with the velocity, a velocity-time diagram in percentage becomes identical with the flow-time curves and therefore could be designated as a velocity-wave curve at these points, in the same manner as the head curves are actually pressure-wave curves at the same sections. For better understanding of the phenomenon, this would be helpful, as the differential of the velocity-time curve gives the acceleration of flow.

The very excellent graphical solution[10] of Professor Bergeron

[4] Professor of Mechanical Engineering, University of Toronto, Toronto, Canada. Fellow A.S.M.E.

[5] "The Action of Valves in Pipes," by R. W. Angus, *Journal*, American Water Works Association, vol. 30, no. 11, Nov., 1938, p. 1858.

[6] Chief Engineer, Byron Jackson Co., Los Angeles, Calif. Mem. A.S.M.E.

[7] "Symposium on Water Hammer," Recommended Standard Symbols, A.S.M.E., 1933, p. 8.

[8] Ref. 3, Fig. 4.

[9] Ref. 3, Figs. 3, 6, 7, and 8; present paper, Figs. 13 and 14.

[10] "Pompes centrifuges et usines elevatoirs," by M. L. Bergeron, *La Technique Moderne*, March 1, 1935.

reproducing the numerical data in a diagram would help further to illustrate the phenomenon step by step.

The author calls attention to the very short but powerful electrical-braking effect of the motor, during the time interval between the opening of the energized circuit at a far-away point and the opening of the control that takes the motor off the line. The importance of this effect in this case is clearly shown and calls for inclusion of this factor in analyzing similar problems.

The publications describing the great project of the aqueduct of the Metropolitan Water District of Southern California, including the author's two papers, show that they blazed new trails in many fields and by giving all details made notable contributions to the art of designing and building of great waterworks.

Ray S. Quick.[11] The author and his associates are to be congratulated upon the excellent test results secured in the control of pressure surge and reverse flow in the discharge lines of the pumping plants on the Colorado River Aqueduct of the Metropolitan Water District of Southern California. The prediction of the surge with respect to valve timing shows a satisfactory agreement with the test results and confirms the engineering principles developed and used in the calculations.

The pumps for the Intake and Gene Plants were manufactured in the San Francisco works of The Pelton Water Wheel Company in collaboration with the Byron Jackson Company. The discharge valves for the Eagle Mountain and Hayfield Pumping Plants were manufactured jointly by The Pelton Water Wheel Company in San Francisco and by the Chapman Valve Manufacturing Company, the cover and operating mechanism having been designed and furnished by Chapman. When it is realized that the completed valve, illustrated in Figs. 6 and 8 of the paper, has an approximate weight of 50,000 lb, the problems of manufacture and handling will be appreciated.

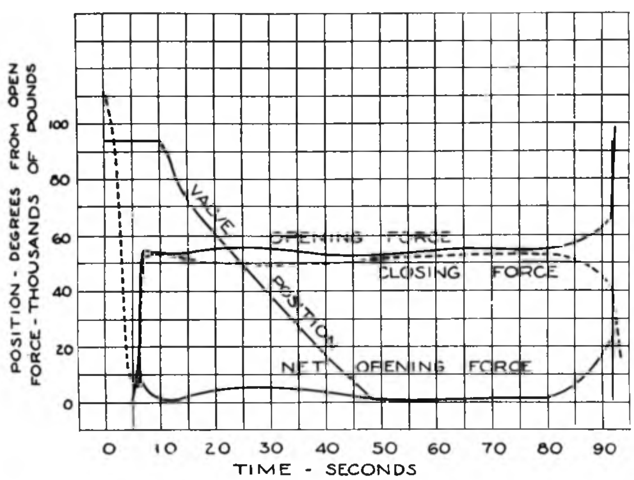


Fig. 15 Normal Opening Cycle of Valve
(Hydraulic cone valve, $40^1/_2$ in. by $53^7/_8$ in.; Eagle Mountain pumping plant.)

It may be of interest to consider further the performance of the discharge valves as tested at the Eagle Mountain Pumping Plant where the degree of control of pressure surge and reverse flow was equally satisfactory to that at Gene as reported by the author. Three sets of curves, Figs. 15, 16, and 17, are submitted to show the history of the valve action with respect to time for the conditions of normal opening, normal shutdown, and emergency shutdown, respectively. Opening, as illustrated in Fig. 15,

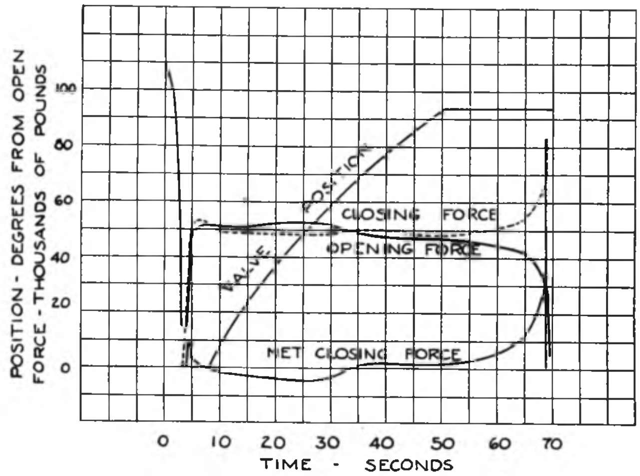


Fig. 16 Normal Closing Cycle of Valve
(Same as Fig. 15.)

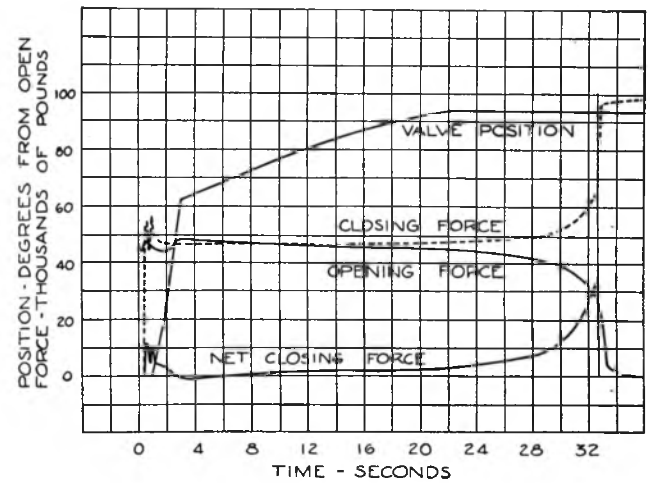


Fig. 17 Emergency Closing Cycle of Valve
(Same as Fig. 15.)

takes place only under normal conditions, with the pump running and with pressures substantially equalized on both sides of the valve. Since the rate of opening is controlled by the actuator unit developed and furnished by the Woodward Governor Company, with timed motor control, the forces on the operating mechanism are moderate and there is little or no pressure disturbances. Likewise, normal closing, as shown in Fig. 16, involves forces of moderate magnitude and little or no pressure disturbance, as here again the control motor is used to produce a steady rate of movement. These two cycles of normal opening and normal closing are used under operating conditions as there is no occasion to bring about emergency closure, except in case of power failure or other abnormal operating condition, requiring rapid shutdown.

The performance, shown in Fig. 17, illustrates the condition of emergency closure. It is interesting to note that, even under these conditions, the forces upon the operating piston are moderate and the reserve available at full pressure leaves a factor of safety of a very satisfactory amount. The greatest force is developed when wedging the plug into the seat in the closed position, as here it is desired to have a watertight closure. In the open position, the plug is very lightly wedged, in order that a wide margin of operat-

[11] Chief Engineer, The Pelton Water Wheel Company, San Francisco, Calif. Mem. A.S.M.E.

ing force may be available for emergency closure. When it is considered that the available force in the operating cylinder is about 115,000 lb, the factor of safety even for wedging is on the order of 3.

Laboratory tests conducted on the model cone valve showed the presence of a definite closing torque with hydraulic flow through the partially closed valve. This closing torque, however, is of relatively small magnitude. A study of the figures confirms this evidence and demonstrates that the greatest load on the operating mechanism occurs in the act of wedging the plug into the seat.

During the tests covered by the foregoing figures, the normal pressure in the discharge line was about 196 lb per sq in. On power interruption, the dip in pressure reached a minimum average of 162 and a maximum average of 207 lb per sq in. Thus the rise in pressure was only 5.5 per cent above the pumping head. These tests are well within desired limits and demonstrate conclusively the effective control of surge.

Cavitation of Hydraulic-Turbine Runners

By R. E. B. SHARP,[1] PHILADELPHIA, PA.

This paper contains a brief analytical discussion of runner cavitation. The I. P. Morris cavitation equipment is described, and results with this apparatus are compared with those of the Holtwood laboratory. A calibration gage in the development stage is also described. Stroboscopic apparatus for photographing runners during operation under cavitating conditions is illustrated, and various photographs above and below the sigma break are reproduced and discussed. A comparison of the index test of a prototype with a model at the same value of sigma is submitted.

THE sigma break of any hydraulic-turbine runner is considered as the value of sigma at which the performance characteristics begin to undergo a change due to cavitation. This value increases with the specific speed of the runner, i.e., the greater the specific speed, the greater, in general, must be the allowance for pressure drop $H_b - H_s$ on the back of the blades in the usual Thoma formula (1).[2]

Sigma $= \dfrac{H_b - H_s}{H}$ with H_b as the height of the water barometer and H_s the vertical distance from the center line of the runner; or more strictly, from the local point of lowest pressure on the blade, to the tailwater elevation. The reason for this increasing value of $H_b - H_s$ with increasing specific speeds is due to several factors, namely, the greater pressure differential necessary on the two sides of the blades, the greater absolute velocity, and greater relative velocity at discharge.

In order to present an approximate comparative picture of the increase in differential blade pressure with increasing specific speed, Fig. 1 and Table 1 have been prepared. Runners of three different speeds have been arbitrarily selected, these values being 50, 80, and 135; the two former speeds falling within the usual range of Francis-type runners, and the latter of propeller runners. The proportions and velocities selected correspond with usual practice, and the method employed at arriving at the approximate pressure differential is based on the efficiencies being normal, without excessive losses. Inflow and outflow diagrams are indicated, these being taken in all cases for the sake of simplicity at the runner throat or periphery with the flow axial, i.e., without radial component. In every instance both the runner diameter and the head acting are considered as 1 ft.

The basic-energy relation as set forth by the Eulerian theorem is

$$u_1 V u_1 - u_2 V u_2 = gHe \quad [1]$$

where u_1 and u_2 = velocity of rotation in feet per second at inflow and outflow, respectively; Vu_1 and Vu_2 = components of the absolute velocity V_1 and V_2 in the direction of rotation; g = acceleration due to gravity; H = head; and e = hydraulic efficiency. With $u_1 = u_2 = u$ and $H = 1$

$$Vu_1 - Vu_2 = \frac{ge}{u} \quad [2]$$

Where $HP = \dfrac{WQe}{550}$, multiplying Equation [2] by $\dfrac{WQu}{g \times 550}$, we have

$$\frac{WQe}{550} = \frac{WQu}{g \times 550}(Vu_1 - Vu_2) = \frac{2\pi FRN}{550} \text{ with } N = \frac{RPM}{60}$$

$$= \frac{u}{\pi D},\ D = 1 \text{ ft},\ R = \frac{D}{2},\ W = 62.4,\ F = \frac{WQ}{g}(Vu_1 - Vu_2)$$

= tangential force acting on blades at radius R.

Consider as Q the discharge through an annulus of Δr radial depth. Then the area on which F is acting per blade is $L \times \Delta r$, Fig. 1 and Table 1. The force F, however, is the tangential component of F_1 which for this comparison is considered as acting on the angle θ with the tangential, this angle being such that the direction of the force F_1 bisects the angle between v_1 and v_2.

The various steps are indicated in Table 1. Item 21 represents the area plotted between L_a and h_d, the differential blade pressure in feet, projected in the direction of rotation in the annulus Δr to produce each of the specific speeds under consideration. In Fig. 1, these areas are shown at the right-hand portion of the figure. Admittedly the shapes are inaccurate, but the areas serve to indicate the intensity of differential pressure to produce the three specific speeds selected. The minimum pressure is plotted as sigma H, these values in each case being about those usually encountered. Sigma H is made up of the terms, $\dfrac{Vm^2}{2g} \times$ draft-tube efficiency plus the term K_cH (2). The expression $\dfrac{Vm^2}{2g} \times ed$ represents the pressure drop due to head regain in the draft tube. The term K_cH represents the local pressure drop on the blades.

The portion of the blade pressure lower than that corresponding with tailwater level or zero on the head scale is a much greater proportion of the total for the propeller-type runner than for the Francis type of lower specific speeds. A region of relatively low pressure has been shown for the backs of the blades at inflow. This is to some extent the result of actual observation of cavitation (for the propeller type) as discussed later, and is also in line with measured values for airfoils. This same tendency, as indicated by pitting, apparently exists to a minor extent on Francis-type runners. The greater preponderance of positive differential pressure in the Francis runners (above 0) with a relatively small amount below the 0 line is the reason for the difficulty of determining by test, with lower specific speeds, the definite location of the sigma break. In other words, loss of head due to cavitation on the backs of the blades has a relatively small effect on the force causing rotation. The intensity of the differential pressure could, of course, be reduced for the propeller runner by increasing the blade length, but to do so would so increase the friction loss that, with the high relative velocities existing, the value of ϕ at which maximum efficiency occurs would be reduced.

Pitot-tube traverses taken just below a propeller-type runner at a steep blade angle demonstrate that the axial velocities are materially lower at the wall of the tube than further inward and indicate the increasingly severe conditions as regards cavitation for some distance in from the periphery of propeller-type runners when operating at the steeper blade angles. This condition has been taken into account in the diagrams of the propeller-type

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[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Hydraulic Division, and presented at the Spring Meeting, Worcester, Mass., May 1–3, 1940, of The American Society of Mechanical Engineers.

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

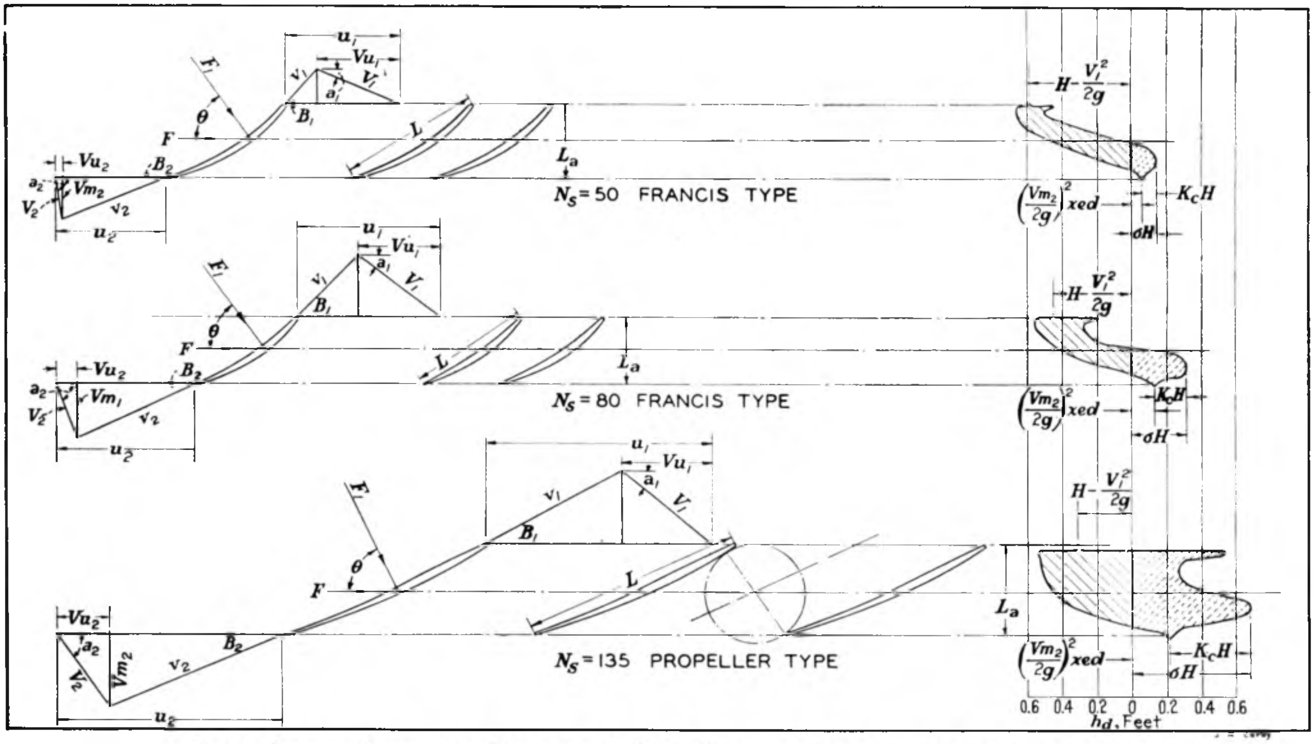

Fig. 1 Showing Increase in Differential Blade Pressure With Increase in Specific Speed

runner in Fig. 1; that is, the value of Vm_2 in that figure is less than that corresponding with the average flow for this runner. Traverses taken at lower blade angles show that this condition of relatively low velocity at the wall of the tube dies out for lower flows. On this account, and in consideration of other test data, the flow for the two lower specific speeds in Fig. 1 has been considered as uniform across the runner discharge in arriving at the differential areas shown.

The increased cavitation tendency of runners of higher specific speed is also aggravated by the increased relative velocity. The pressure existing at any point on the blade where water tends to leave the surface is a function of the square of the relative velocity. Fig. 1 indicates by the comparative values of v_1 and v_2 for the different specific speeds how much greater this sensitivity is with the increased specific speeds, with resulting need for finishing of the blades true to contour.

That portion of each area, in Fig. 1, representing intensity of differential blade pressure which is below 0 on the head scale in the figure, is a function of Vm_2 and v_2. These values, as has been pointed out, increase with specific speed; and in addition, the ratio of the maximum to the average values of Vm_2 and v_2 undergoes an increase. As a consequence, certain portions of the blade may have a value of sigma H appreciably greater than the sigma H as determined by the change in performance characteristics. This is true even with runners having sharp and well-defined sigma-break curves.

It is not economically practicable to have the margin between the sigma break and the plant sigma sufficient to insure the absence of cavitation on the entire blade surface on this account. It is, therefore, becoming common practice to preweld portions of the runner blades with stainless steel where the operating head is above about 50 ft, in addition to providing a margin between the sigma break and the plant sigma (3).

Fig. 2 is a view of a turbine-runner blade before being welded. It will be noted that the portion prepared for welding extends along the entire periphery with a greater portion near the inflow

TABLE 1 DATA ON RUNNERS USED TO COMPARE INCREASE IN DIFFERENTIAL BLADE PRESSURE WITH INCREASE IN SPECIFIC SPEED

Item[a]	Francis type		Propeller type
(1) $N_s = RPM \sqrt{\frac{HP}{H^{5/4}}}$	50	80	135
(2) $\phi_1 = \phi_2$	0.8	1	1.6
(3) $u_1 = u_2 = \phi\sqrt{2gH}$, where H = 1 ft, fps	6.42	8.02	12.83
(4) a_2 = absolute angle discharging water makes with rotation, deg	80.5	69.5	54
(5) Vu_2 = tangential component at discharge, fps	0.4	1.2	3
(6) $Vu_1 = \frac{gHe}{u} + Vu_2$, fps	4.66	4.61	5.13
(7) $Vu_1 - Vu_2$, fps	4.26	3.41	2.13
(8) Vm_2 = axial component of discharge, fps	2.32	3.25	4.2
(9) Area of annulus, of Δr radial depth with radius r = 6 in., 7 per cent deducted for blade area at discharge, sq ft	$0.244\,\Delta r$	$0.244\,\Delta r$	$0.244\,\Delta r$
(10) Quantity flowing through annulus = (9) × (8), cfs	$0.566\,\Delta r$	$0.793\,\Delta r$	$1.025\,\Delta r$
(11) $F = \frac{WQ}{g}(Vu_1 - Vu_2) = \frac{W}{g}(10) \times (7)$, lb	$4.68\,\Delta r$	$5.24\,\Delta r$	$4.25\,\Delta r$
(12) B_1 = relative angle at entrance, deg	50	46.5	29
(13) B_2 = relative angle at discharge, deg	21	25.5	23
(14) $\frac{(180 - B_1 + B_2)}{2} - B_2 = \theta$, deg	54.5	54	64
(15) $\frac{F}{\cos\theta} = F_1 = \frac{(11)}{\cos\theta}$, lb	$8.05\,\Delta r$	$8.92\,\Delta r$	$9.7\,\Delta r$
(16) Number runner blades = N	17	16	5
(17) L = distance inflow to outflow edge, in.	4	3.445	6.29
(18) L_a = axial depth of blades, in.	2.125	1.84	2.53
(19) F_1 per blade = $\frac{(15)}{(16)}$, lb	$0.474\,\Delta r$	$0.557\,\Delta r$	$1.94\,\Delta r$
(F_1 also = $\frac{\Delta r(L \times h_d)}{2.304}$ where h_d = differential blade pressure in ft of water per sq in.)			
(20) $L \times h_d = \frac{F_1 \times 2.304}{\Delta r}$ = area plotted between L and h_d, sq in.	1.09	1.28	4.47
(21) (20) × $\frac{L_a}{L}$ = area projected in direction of rotation, sq in.	0.58	0.685	1.8

[a] Numbers in parentheses in this table are key numbers not to be confused with Bibliography.

edge, and also near the outflow edge. In addition, a strip of rolled stainless steel is welded to the periphery of the blade, the need for which is discussed later. The portion of the blade which is to be prewelded is cast with a depression about $^1/_4$ in. deep, and stainless steel containing essentially 18 per cent chromium, 8 per cent nickel, and with the carbon limited to about 0.08 per cent (3), is welded as indicated, precautions being taken to prevent distortion of the blade during this process.

Fig. 2 Turbine-Runner Blade Before Being Prewelded

Cavitation Laboratory

The need for the convenient and prompt determination of the cavitation characteristics of model propeller runners resulted in the construction of the I. P. Morris cavitation equipment (5) in 1938, at Eddystone, Pa., Fig. 3. This involves essentially a closed circulating system with a service pump located about 8 ft below the floor, thus permitting relatively high values of H_s in the turbine draft tube without having as high values at the pump. A calibrated venturi meter measures the discharge. The tests are run under heads up to 35 ft. This equipment is designed primarily for tests of propeller runners of 11 in. throat diam, although it is possible in the case of Francis-type runners to increase this dimension. It is also designed for the cavitation testing of pumps. Variations of sigma are obtained by changing the pressure throughout the system (see Appendix for engineering data and equipment pertaining to cavitation equipment).

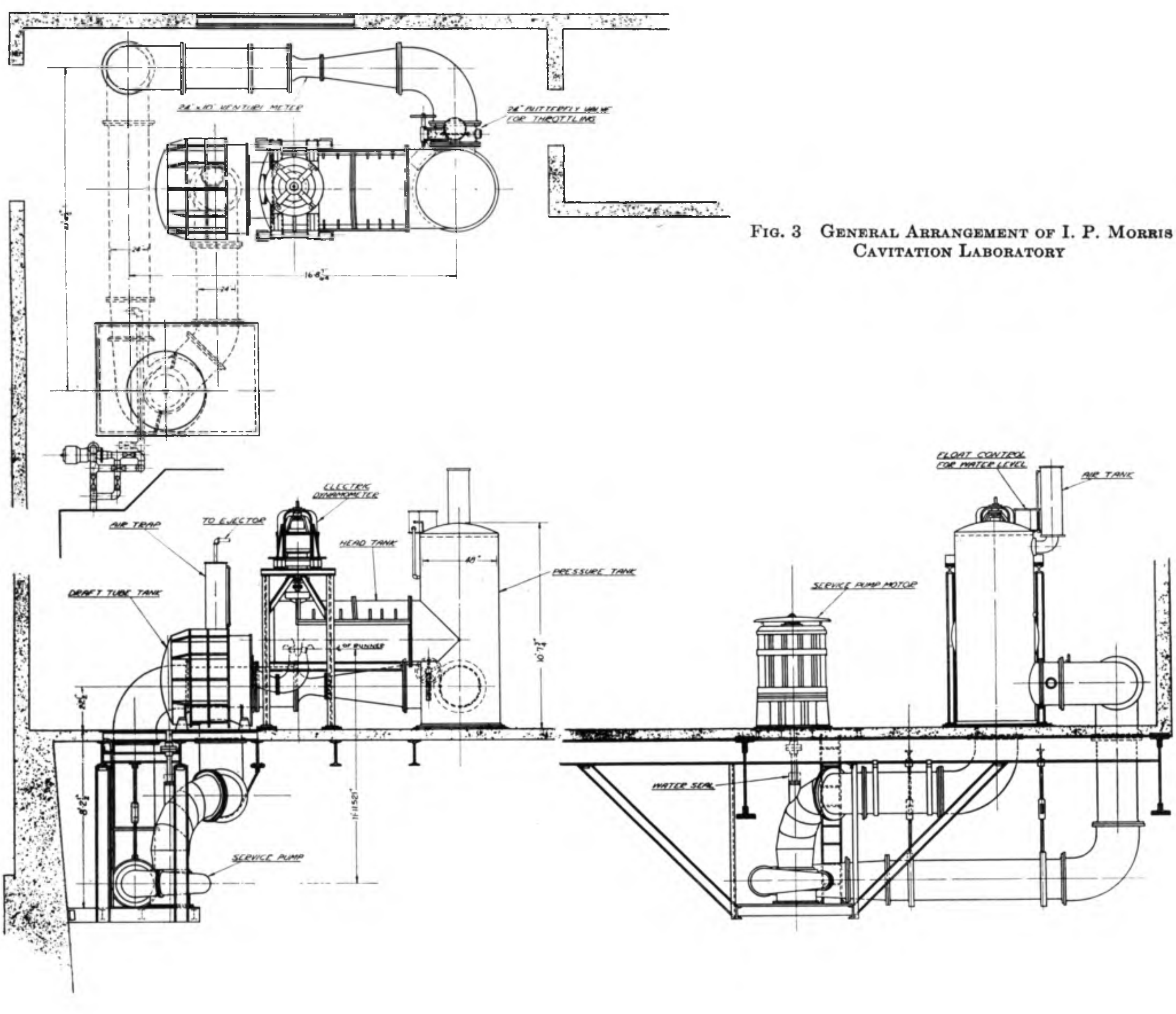

Fig. 3 General Arrangement of I. P. Morris Cavitation Laboratory

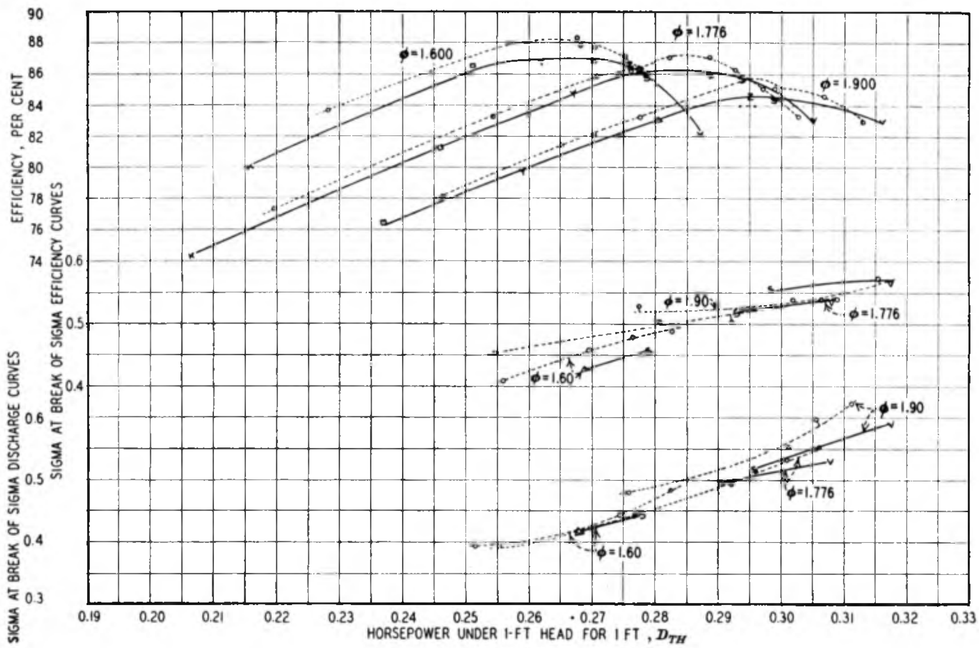

Fig. 4 Comparison of I. P. Morris Laboratory 11-In. Runner Model With Holtwood Laboratory 16-In. Prototype

This equipment was designed to permit the air to be separated from the water as much as possible, with the recognition that an undue air content would affect the sigma-break values (3, 4). The vertical pressure tank permits the air to rise to the top where it may be drawn off. However, it was later found advisable to add the air trap in the suction tank, and this has been the most effective means of air elimination since most of the air separation takes place in the draft tube. Particular attention has been paid to the possibility of air leaks in the suction tank and at the stuffing box to the service pump, where a water seal has been provided.

To calibrate this equipment as to efficiencies obtained and as to sigma-break values, a comparison was made with the Holtwood laboratory of the Safe Harbor Water Power Corporation by constructing and testing an exactly homologous but smaller (11-in. runner diam) model, for comparison with the Holtwood test of a 16-in. similar model. This particular 11-in. model is used as a standard at the I. P. Morris laboratory for check tests from time to time. Fig. 4 shows the results of a recent (March, 1940) test as compared with the Holtwood test made during June, 1935. The lower efficiencies at Eddystone are consistent with the smaller runner diameter, and the power and shape of the performance curves check closely.

The lower portion of Fig. 4 shows the comparison of sigma breaks for three values of ϕ. The upper set of curves is based on the break in efficiency and the lower set on quantity. While there is somewhat of a difference at 1.9ϕ, particularly on the quantity basis, the remaining curves check fairly well. At least, there is no systematic tendency for the 11-in. Eddystone results to show a higher sigma break due to air content. The tests were made at heads from 27 to 30 ft at Eddystone and under heads of 40 to 60 ft at Holtwood. It has subsequently been found necessary to keep above the 25-ft head at Eddystone to avoid an increase in sigma break due to air content. It is probable that this lower limit could be reduced if this were a noncirculating system.

In an effort to provide a means for determining whether or not or to what extent cavitation exists in a given turbine without recourse to measurement of change of characteristics, the device shown in Fig. 5 has been tried, with partial success. This device as tried out up to this time has been installed in the throat ring of

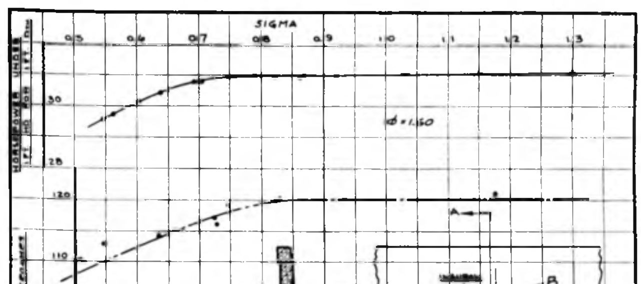

Fig. 5 Comparison of Sigma-Break Curves With Cavitation-Gage Reading

(Sigma curves for ϕ = 1.60 and 1.80.)

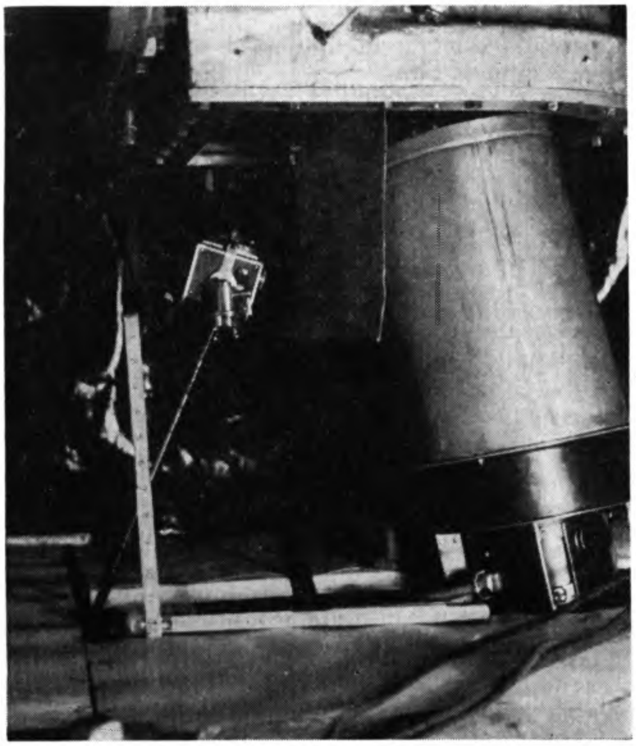

FIG. 6 STROBOSCOPIC EQUIPMENT FOR STUDYING CAVITATION

the turbine. The functioning of the device is based on the change in resistance between the points A and B to an electric current, when operating in cavitation-free water as compared to water with cavitation present. The change in current, in comparison with the measured change in characteristics is indicated in this figure. For $\phi = 1.61$, it may be noted that the change in resistance takes place at about the same value of sigma as that at which the power developed and discharge start to change; whereas, at the higher value of ϕ (1.805), the performance characteristics undergo a change somewhat prior to the change in resistance. A gage of this sort may have possibilities in view of the ease in obtaining readings at any time during the operation of the turbine.

PHOTOGRAPHIC RECORD OF RUNNER TESTS

With a view of having the report of each runner tested include photographs of the location and progress of cavitation for decreasing values of sigma, at different values of ϕ stroboscopic equipment, as indicated in Fig. 6, has been provided in connection with the I. P. Morris cavitation equipment. This too is yet in a state of development.

The elbow portion of the draft tube has been provided with an opening covered with "Plexiglass" which conforms with the curved portion of the tube. Below this is a flat glass opening, the space between these parts being filled with water during operation to eliminate distortion of vision. An Edgerton stroboscope is located on one side of a partition below this opening, with the other side of the partition used for visual study or for photography.

The stroboscope is wired to a commutator geared to the turbine

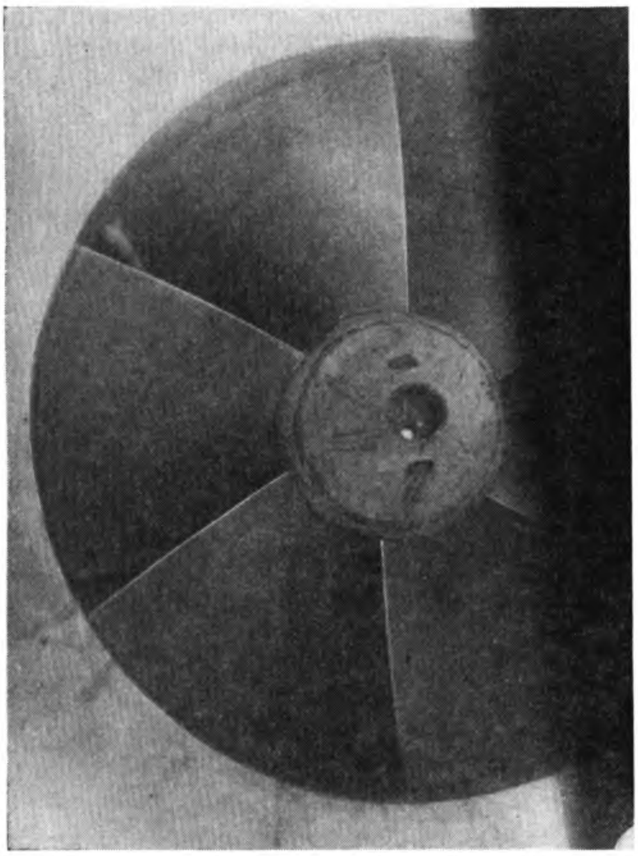

FIG. 7 STROBOSCOPIC VIEW OF PROPELLER-TYPE RUNNER (MODEL 128) WITH FIXED BLADES

(Test No. 617; water temperature = 68 F; head = 32.32 ft; h_s = —0.94 ft; $0.8H_v$ = 5.15; sigma = 1.055; HP_1 = 0.272; Q_1 = 2.76; efficiency = 86.8 per cent; ϕ = 1.60; break at sigma = 0.44.)

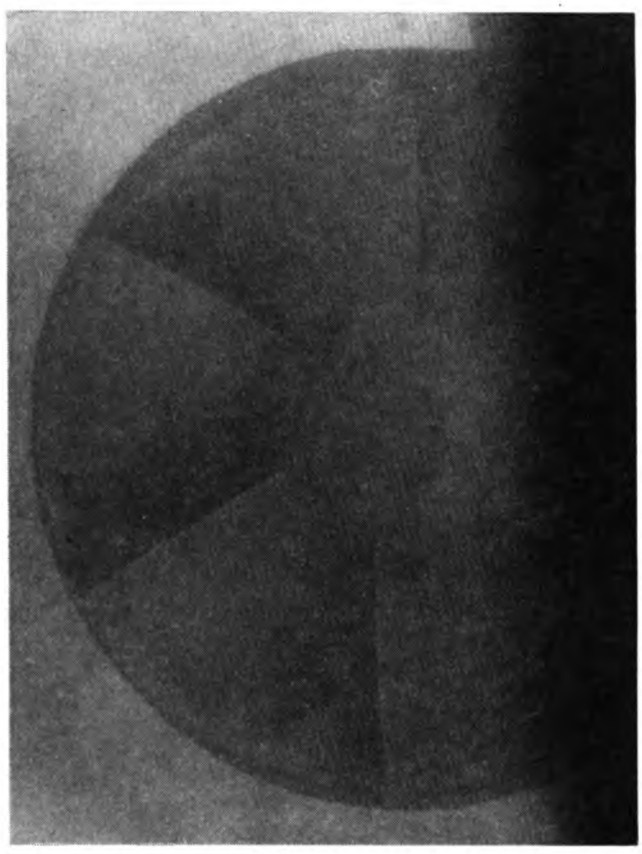

FIG. 8 STROBOSCOPIC VIEW OF PROPELLER-TYPE RUNNER WITH FIXED BLADES WHEN OPERATING JUST ABOVE SIGMA BREAK

(Test No. 617; water temperatures = 68 F; head = 32.22 ft; H_s = +16.21 ft; 0.8 H_v = 5.10; sigma = 0.525; HP_1 = 0.271; Q_1 = 2.75; efficiency = 87 per cent; ϕ = 1.60; break at sigma = 0.44.)

shaft and normally flashes once at each revolution, thus giving a view of each separate blade rather than a composite effect. Time exposures have been found necessary, even with the most sensitive film available, due to the limited amount of light possible for photography. An alternate method has been used sometimes whereby photographs with single flashes were taken, requiring a special condenser for increasing intensity. A lens is provided above the stroboscope to confine light rays as much as possible to the runner blades. The photographs taken with sigma values well above the break, i.e., with relatively high pressures in the draft tube, have been quite successful. However, as the sigma is lowered with consequent reduced draft-tube pressure, the water below the runner becomes cloudy due to air coming out of solution, with resulting interference to the photography. Unfortunately, the photographs reproduced in this paper were taken during the winter months, and it was not practicable to raise the water temperature and thus reduce the amount of air in solution. It is expected that photographs taken during the summer months with water temperatures as high as 85 F will show better results.

Figs. 7 and 8 are stroboscopic views of a propeller-type model runner 128 with fixed blades and with the periphery of the blades turned cylindrical to give minimum clearance throughout their entire length inside of the cylindrical throat ring.

Both photographs were made at ϕ = approximately 1.60. Although best efficiency is at $\phi = 1.55$, it will be noted from Fig. 7 that cavitation starts at the intake edges on the backs of the blades near the periphery, at sigma values considerably higher than the break. This condition is doubtless aggravated at $\phi = 1.55$ where efficiency is maximum. The obvious suggestion for the correction of this condition is to design the blades with a steeper angle of intake where the cavitation takes place. This steeper angle results in excessive blade curvature, and reduces the efficiency and power at higher values of ϕ, which is quite undesirable, particularly where heads below normal are encountered. The efficiency performance of this runner, the unusually low and sharp sigma break, and the occurrence of cavitation at this point on many other runners leads to the belief that this may be a normal condition. This is also the point of minimum pressure on efficient airfoils (8).

Fig. 8 gives a very indistinct idea, due to air separation, of conditions as the sigma break is approached. With the sigma break at 0.44, Fig. 8 indicates the cavitation extending along the blade but still with a greater amount at inflow than at outflow. Some cavitation has appeared near the hub. It may be noted that in neither of these illustrations is there any indication of cavitation at the periphery due to flow through the clearance spaces. This is in sharp contrast with conditions indicated later with Kaplan-type runners, where the clearance at most of the periphery is necessarily greater.

Due to the manner in which the water clouds in these photographs, preventing a clear view of extensive cavitating conditions, it is planned to redesign the equipment in the I. P. Morris laboratory by lowering the turbine so that, by having a portion of the throat ring and upper draft tube transparent, the light can be introduced in a radial direction from the side, with observations and photographs made from adjacent points. This will permit observations of the face side of the blades, with a much shorter distance for the light to travel. The photographs show, however,

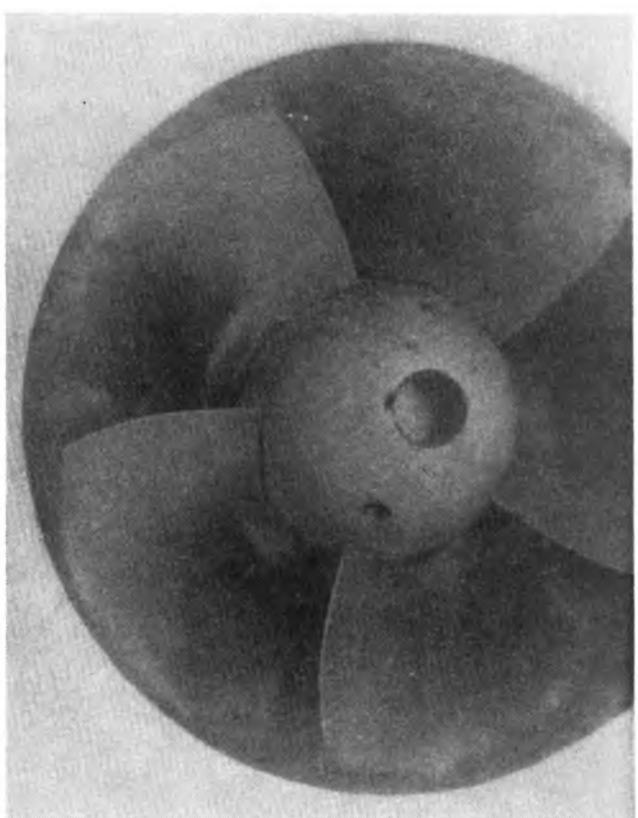

FIG. 9 KAPLAN-TYPE MODEL RUNNER 142-D OPERATING AT SIGMA BREAK
(Test No. 616-II; water temperature = 60 F; head = 30.67 ft; $H_s = +8.11$ ft; $0.8\,H_v = 8.27$; sigma = 0.815; $HP_1 = 0.33$; $Q_1 = 3.53$; efficiency = 82.3 per cent; $\phi = 1.60$; break at sigma = 0.81.)

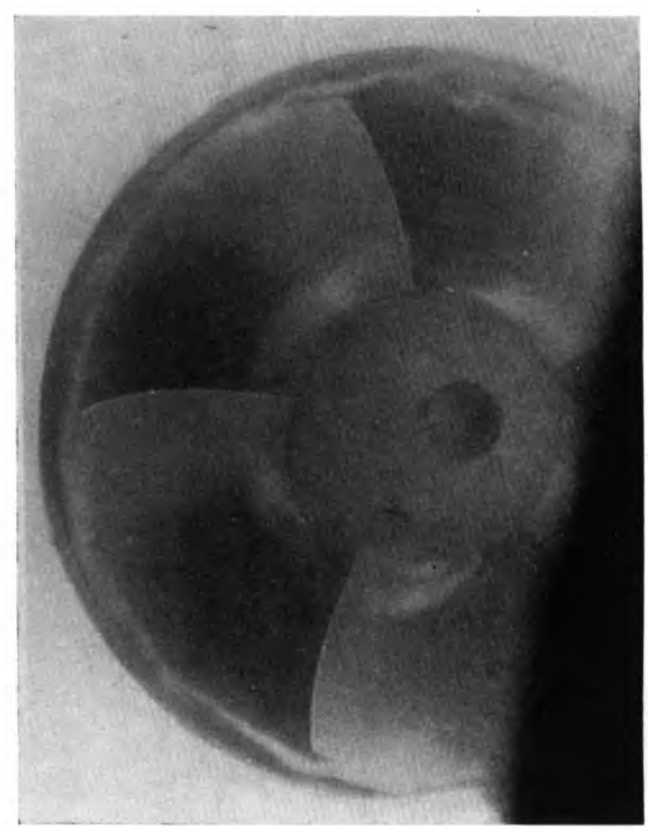

FIG. 10 KAPLAN-TYPE MODEL RUNNER 142-D OPERATING AT HIGHER SPEED AND BEYOND SIGMA BREAK
(Test No. 616; water temperature = 65 F; head = 29.5 ft; $H_s = +4.16$ ft; sigma = 0.972; $0.8\,Hv = 10.55$; $HP_1 = 0.361$; $Q_1 = 4.06$; efficiency = 78.6 per cent; $\phi = 1.80$; break at sigma = 1.05.)

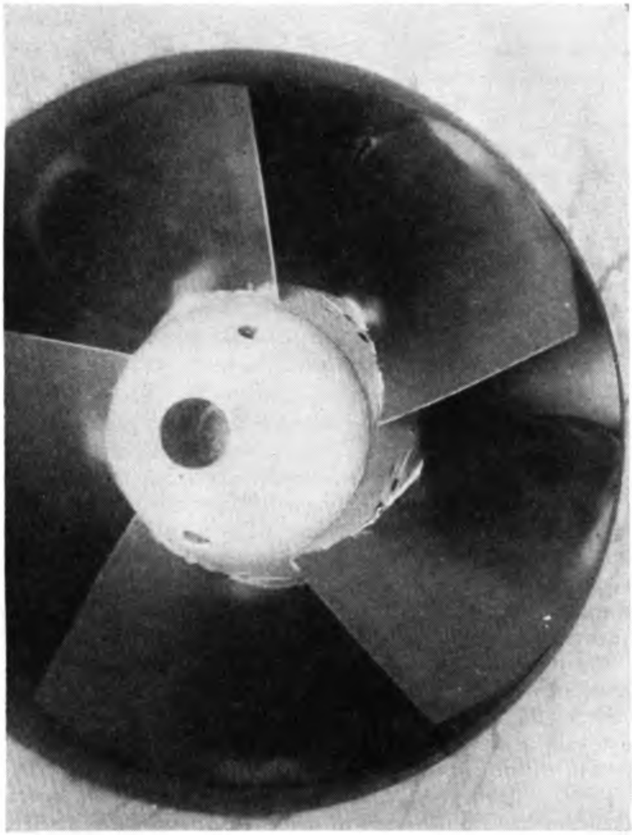

FIG. 11 KAPLAN-TYPE MODEL RUNNER 148 AT SIGMA BREAK, SHOWING CAVITATION AT HUB AND AT PERIPHERY

(Test No. 618; water temperature = 63 F; head = 29.74 ft; H_s = —6.04 ft; sigma = 1.307; HP_1 = 0.359; Q_1 = 3.97; efficiency = 79.8 per cent; ϕ = 1.60; break at sigma = 1.30.)

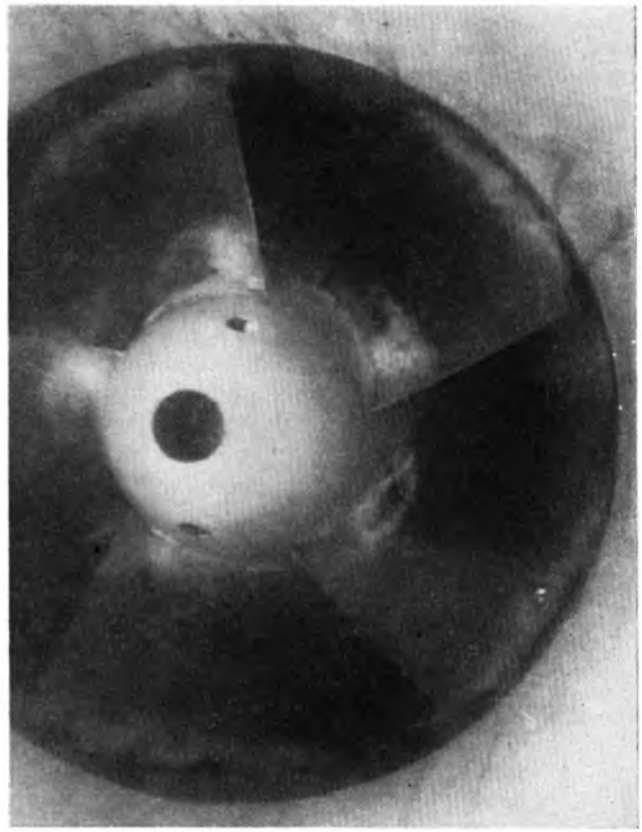

FIG. 12 KAPLAN-TYPE RUNNER MODEL 148, OPERATING FAR BEYOND SIGMA BREAK

(Test No. 618; water temperature 63 F; head = 30.5 ft; H_s = +8.06 ft; sigma = 0.817; HP_1 = 0.330; Q_1 = 3.80; efficiency = 76.6 per cent; ϕ = 1.60; break at sigma = 1.30.)

where cavitation is initiated and this is important as indicating the location of faults in design, or the desired location for prewelding.

Fig. 9 is a view of model No. 142-D which is of the Kaplan type with spherical throat ring. The value ϕ being 1.60 is slightly above that at best efficiency. Sigma is 0.814, the break occurring at 0.81. Cavitation is occurring at the periphery, at the intake, and through the clearance space just upstream and downstream from the blade axis where clearance is a minimum. Cavitation near the hub indicates rather too abrupt blade curvature, about at the blade axis.

At $\phi = 1.8$, Fig. 10, sigma 0.972, break 1.05, the disturbance at the hub is accentuated, with indications of outward radial component of flow due to the higher whirl component Vu_2. The cavitation in the clearance space is also aggravated with streamers extending some distance beyond the ends of the blades. Punishment of the throat ring at the blade discharge is indicated. This latter condition exists to a lesser degree at this ϕ (1.8) well above the sigma break.

Figs. 11 and 12 show model No. 148 at $\phi = 1.60$ with sigma = 1.307, 0.817, respectively, where the sigma break is at 1.3. The marked increase in cavitation at the hub will be noted. The entire series of photographs taken of this runner at $\phi = 1.4$, 1.6, and 1.8 shows an interesting feature, namely, that cavitation at the periphery develops prior to the break in sigma, particularly at low ϕ values. Cavitation near the hub tends to develop later and, at least on this runner, it is the cavitation at this point rather than at the periphery which affects the performance by reducing the output and efficiency. It, therefore, appears that this is the portion of this runner which should be altered for improvement. In general, these studies show the importance of the blade design near the hub, also the need for smoother hub surface between the blades. Cavitation near the hub, however, seldom if ever results in pitting of the prototype due to the low relative velocity v, or more improbably, to the voids at this point on the runner not collapsing on the runner-blade surface.

Model No. 148 was made exactly similar to a former model with the exception that the blade axis was moved further downstream on the blade. The same blade pattern and templates were used, but with a shift in the axis. This change has had the effect of improving the efficiency and sigma break. The effect of this change is to reduce the clearance between the throat ring and the outflow portion of the blade, and the change is consistent with the improvement found with a properly designed spherical throat ring as compared to a cylindrical throat ring.

It is believed that the proper interpretation of stroboscopic photographs of runners is a valuable means of eliminating faults in design. The relation between the progressive change in cavitation with decreasing sigma to the change in performance can be used in deciding on the changes to be made for expected improvement. However, a liberal number of photographs showing the progressive changes are essential. By constructing the blades of a tough, ductile bronze, the same ones may be used for a large number of different blade shapes. Plaster of Paris forms may be kept as a record for each change made. Accurate templates are, of course, necessary for each change, to insure all blades being uniform for each test made.

In connection with the disturbance in the clearance space, Fig.

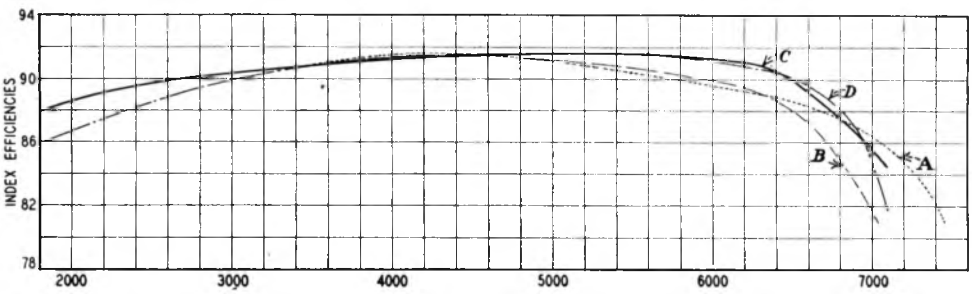

Fig. 14 Comparison of Sigma Break on Model With That on Prototype

(A = 16-in. model tested with high sigma; B = 11-in. model tested with sigma same as prototype; C = 109.5-in. prototype unit No. 1; D = 109.5-in. prototype unit No. 2.)

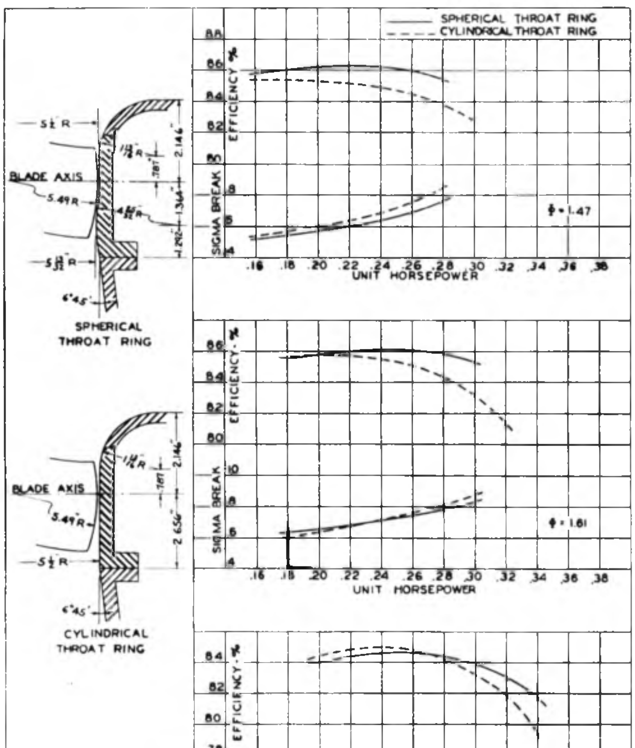

Fig. 13 Curves Showing Relative Cavitation and Efficiency Performance of Kaplan-Type Runners With Spherical Throat Ring and Straight Cylindrical Throat Ring, Respectively

13 is introduced to show the relative performance (6, 7) as to efficiency and cavitation of a Kaplan-type runner (*a*) with spherical throat ring the same as used previously, and (*b*) with straight cylindrical throat ring. The former results in less clearance space disturbance but puts a greater burden on the draft tube due to the smaller minimum diameter of the throat ring. The latter is important when it is remembered that draft-tube losses vary with the fourth power thereof. The improved results with the smaller clearance space emphasize the importance of the spherical shape, if properly designed.

Sigma Break on Model and Prototype

Fig. 14 is presented as an example of the relation between sigma break on the model and on the prototype. In this instance the model is homologous to the prototype throughout, including intake, casing, and draft tube. The prototype was not tested as to efficiency but the power was recorded accurately, and an index test was made by taking readings of Winter-Kennedy piezometers in the casing and stay ring. The maximum efficiencies shown are those stepped from the 16-in. model runner to a diameter of 109.5 in. by the Moody formula (10). Those of the 11-in. model have been arbitrarily stepped up to give the same maximum value as have those of the prototype.

The 16-in-model test was made at high sigma values, and is, therefore, not affected by cavitation. The 11-in-model test was made at the same sigma as the prototype. Some increase in prototype power is noted over the 11-in. model, with fair agreement in the form of the curves.

The flash photograph of runner No. 148, taken with a special condenser which is necessary to increase the intensity of the light to permit such a photograph, is shown in Fig. 15. This is at blade

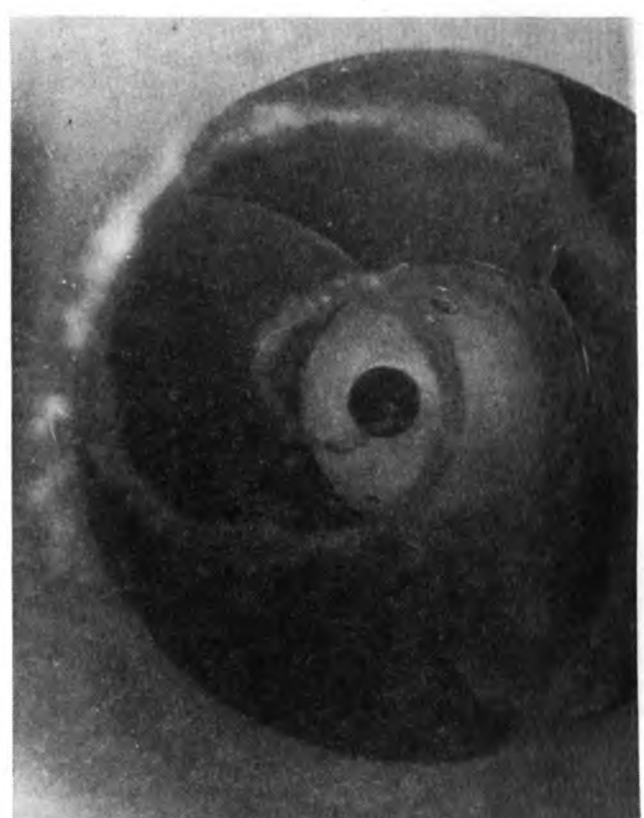

Fig 15 Flash Photograph of Model Runner No. 148 Showing Spiral Vortex

position No. 3 with $\phi = 1.595$, $HP_1 = 1.722$, $Q_1 = 1.82$, efficiency $= 86.0$, and $\sigma = 0.648$. Sigma break under this condition occurs at 0.4.

This vortex, it will be noted, appears to form in the hole in the center of the hub and is rotating with the runner. A time exposure taken under this condition would merely show a blurred effect even with the stroboscope, due to the rotation of the spiral vortex not being in synchronism with that of the runner. As the σ was reduced to a value nearer the break, this vortex disappeared. The gate opening at which this photograph was taken is such that this would represent an actual operating condition at this value of ϕ when the load on the turbine is reduced to slightly more than half of the normal value.

Acknowledgment is made of the helpful suggestions of L. F. Moody in the preparation of this paper, of the assistance of R. B. Willi, and of K. W. Beattie in connection with the photographs and test data included.

Appendix

I. P. MORRIS CAVITATION LABORATORY

General Data and Equipment

Diameter of runners tested: 11 in.

Head on turbine: 25 to 35 ft.

Variation of sigma: Accomplished by changing pressure in system upstream from turbine from below atmospheric to 20 psi above atmospheric.

Water supply: Closed circulating system with cold water connection automatically added from city main as make-up, and to regulate temperature. Volume of water in system about 500 cu ft.

Service pump: Designed for 25 cfs at 35-ft head; driven by 125-hp slip-ring motor, 220 v, 580 rpm rated speed. Possible speed regulation to one-half normal in eleven steps.

Water measurement: 24-in. by 10-in. venturi meter; calibrated by Professor Pardoe at the University of Pennsylvania.

Dynamometer: Specially built, using 40-hp 1200-rpm d-c motor reinforced for higher speeds. Dynamometer carried by cradle mounted on ball bearings. Torque measured by a Fairbanks beam scale connected through flexible steel tape to dynamometer frame. Electrical output dissipated in heat.

Pressure tank: 48 in. diam, 10 ft high, with supplementary air tank at top. Water level in this tank automatically controlled by float valve.

Head tank: 48 in. diam with float bottom, 7 ft 6 in. long.

Draft-tube discharge tank: 54 in. diam, 4 ft long, with trap for collection and drawing off air.

Gages and instruments: Head and water measurement pot-type mercury gages; rpm dial revolution counter, with electric release controlled by clock; atmospheric-pressure mercury barometer checked at intervals from weather bureau; instrument for measurement of air content in water, as developed at Massachusetts Institute of Technology.

Stroboscope: "Strobulux" controlled by "Strobotac," type 648-A, manufactured by General Radio Corporation.

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Combined Tension-Torsion Tests on a 0.35 Per Cent Carbon Steel

By E. A. DAVIS,[1] SO. PHILADELPHIA, PA.

This paper contains the results of a group of combined tension-torsion tests on both cylindrical and notched bars. The author has tried to keep in mind the close relation between this type of test and the problem of bolting. The test results show the amount of reduction in the ultimate tensile strength when a torque is applied in combination with the tension load. They also show a similar reduction in the torsional strength when a tensile load is added.

In general practice, it frequently occurs that some members of a structure or machine carry loads in both tension and torsion simultaneously. It is pretty well known or it may be logically deduced that the addition of a torque to a bar already stressed in tension will reduce the tensile load-carrying capacity of the bar. The amount that the tensile strength will be reduced, however, is not so well known, and engineers in general have tried to avoid this type of loading wherever either the tension or the torsional stress is likely to be high.

A typical example of this combined tension-and-torsion loading is a bolt which has been tightened by a wrench. As the nut is being drawn up, the twisting moment of the wrench is applied to the bolt and there is a danger that the bolt may be strained in torsion to the extent that it will not support its load in tension. Because of this danger, methods have been developed for reducing this twisting moment in the tightening of large bolts. It is quite common practice in the assembly of large bolted parts to expand the bolts by heating to such an extent that the nuts can be turned up just enough to give the bolts their proper working stress when the bolts are at the same temperature as their surroundings.

In some large alternating-current generators, the rotors are made up of several forged disks which are bolted together with long axial bolts. These bolts are drawn up to a given load by a "bolt puller" and the nuts are tightened by hand.

Unfortunately not all bolts can be tightened in this manner and often it is necessary to draw them up with a wrench. With this in mind, the present investigation was carried out in an effort to determine just what effect a superimposed load in one direction has on the strength of a bar carrying a second load in another direction. In order to do this, several sets of combined tension-torsion tests were made. In one set, various fixed amounts of tensile load were superimposed upon constant-strain-rate tests in torsion. This set contained one pure torsion test (zero tension). In another set, fixed amounts of torque were maintained while the bars were pulled at a constant strain rate in tension. In this instance, one of the tests was a pure tension test (zero torque). Another test was run in which the strain rates in both tension and torsion were kept constant. Since these tests are closely related to the problem of bolting, it was felt that the effect of notches and threads should be investigated in conjunction with these combined stress tests. Consequently, some of the foregoing tests were repeated on bars containing circumferential notches.

Theoretical Considerations

Probably the best approach to an understanding of the combined tension-torsion test would be to express the test results in terms of shearing stress and strain on some preferred set of planes in the material. For theoretical reasons, the best results would likely be obtained if the octahedral planes were chosen for making this analysis. These planes are the faces of a regular octahedron which is oriented so that its apexes point in the directions of the principal stresses. A. Nádai (1)[2] has made such an analysis of a combined tension-torsion test on a plastic material having a uniform yield stress $\sigma = \sigma_0 =$ constant. The stress distribution under combined tension and torsion in a round bar is reproduced in Fig. 1, in which it is assumed that the material flows plastically over the entire cross section of the bar.

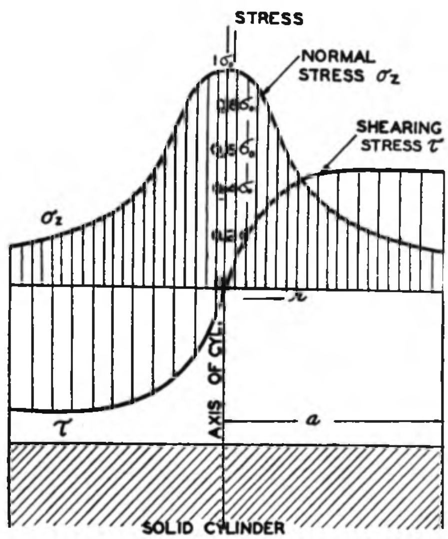

Fig. 1 Stress Distribution Under Plastic Flow in Cylindrical Bar Subjected to Combined Action of Axial Load and Twisting Moment

The distribution of the shearing stresses τ and of the normal stresses σ acting in the axial direction are plotted separately against the radius of the bar. It is to be noted that in the axis of the bar, a state of pure tension exists so that at $r = 0$ the stress in the axial direction is equal to σ_0, the yield stress of the metal in pure tension. Under these conditions, the tensile stress is confined mostly to the inner fibers, while the torque stress is carried by the outside fibers.

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[1] Land Turbine Engineering, Westinghouse Electric & Manufacturing Company.

Contributed by the Iron and Steel Division and presented at the Spring Meeting, Worcester, Mass., May 1–3, 1940, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

When one tries to apply a similar analysis to a solid round bar of a material which exhibits the property of strain-hardening, certain difficulties arise which make an analysis complicated. With a solid bar in torsion the stress-strain diagram is not directly obtained. The shearing-stress (τ) versus the shearing-strain (γ) diagram is obtained from the moment (M) versus angle of twist (θ) curve by the following relations (2)

$$\tau = \frac{1}{2\pi a^3}\left(\theta \frac{dM}{d\theta} + 3M\right)$$

$$\gamma = a\theta$$

This stress and strain refer to the outside fibers of a bar of radius a. The expression for τ involves the slope of the moment-angle curve and this may lead to difficulty in determining the shearing-stress distribution in a test where the applied moment is held constant.

When tension and torsion are to be compared, another difficulty arises from the fact that the expressions for the strains become

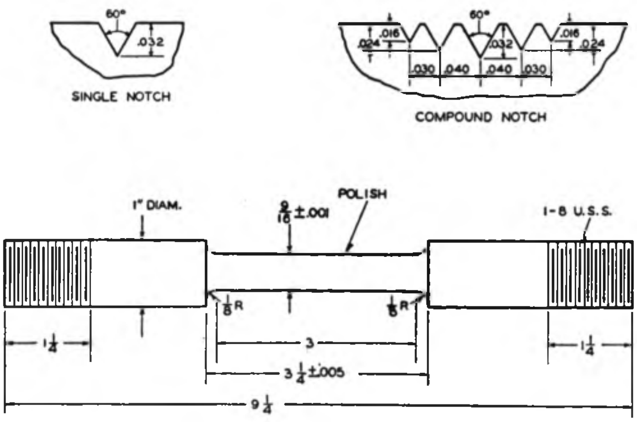


Fig. 2 Tension-Torsion Specimen

complicated if the strains are large. This is especially true for the strains in the octahedral planes. Also, when a bar extends axially, the diameter decreases in order to satisfy the condition that the volume remain constant. Furthermore, since the shearing stress on the outside fibers depends upon the third power of the radius, a change in the stress will occur even though the applied moment is constant. Because of these conditions, the test results in this paper are given as diagrams in which the tensile stress is plotted as a function of the axial strain and the moment of the applied torque as a function of the angle of twist. The tensile stress that is plotted and referred to throughout the paper is that based upon the original area of the bar $\left(\sigma = \dfrac{P}{A_c}\right)$.

The tension-torsion test is an example of a state of combined stress, a field in which a considerable amount of work has recently been done. The aim of most of this work has been to check the general conditions under which the ductile metals yield plastically. Among the investigations of this nature which have been made, those by Lode (3) and by Schmidt (4) in Germany; by Taylor and Quinney (5) and by Cook (6) in England; and by Ros and Eichinger (7) in Switzerland may be of interest. A survey of the results of most of these tests was made by Nádai (8).

The author in a former paper (9) attempted to show the relation between tension and torsion tests on a solid bar of copper by expressing the results of both in terms of the shearing stresses and strains on the octahedral planes.

The agreement was good in the region where the strains were small but, for strains greater than those corresponding to about 15 per cent elongation in tension, the curves for the different tests separated somewhat.

The present group of tests differs from most of the previous combined stress tests in that the stress-strain diagrams were obtained up to the point of rupture, while the others were concerned chiefly with the region in initial yielding. Also, in these tests the equipment was so arranged that either the tension or the torsion load could be held constant while the other was allowed to vary to suit the conditions imposed by a constant strain rate.

Materials and Test Specimens

In choosing a material for these combined tension-torsion tests, it was felt that certain characteristics would be desirable. If the results were to throw any light on the bolting problem the material should be similar to that generally used in bolts. It should be tough and not excessively ductile; however, a very strong steel would require that the dimensions of the test bar be small due to the limited capacity of the testing machine. Furthermore, the elongation associated with the yield-point region should not be too great, for this phenomenon is not observed to any appreciable extent in the alloy steels commonly used in large bolts. As a compromise among these requirements, a medium-carbon steel (0.35 C) was selected for the tests. With this material a bar with a $^9/_{16}$-in-diam test section could be used in the available equipment.

A sketch of the test specimen is shown in Fig. 2. The 1-in-diam heads were used in order that reasonable torque could be applied through the threaded ends without too greatly tightening the threads in the grips. It was realized that, if the axial extension and the angle of twist were to be obtained for strains up to the ultimate strength, the extensometer would have to be clamped on the shoulders of the bar. This has the weakness that the active gage length is somewhat unknown. For purposes of comparison, however, the actual gage length does not need to be known exactly. In these tests, it has been assumed that the gage length in each case was 3.1 in.

It was felt that it would be desirable to compare the results of tests on bars containing notches and threads with those of the cylindrical bars. Two types of notches, a single and a compound notch as shown in Fig. 2, were designed. The compound notch was included in order to determine whether or not an adjoining notch had any tendency to relieve the stress concentration in the main notch.

As a supplement to the combined stress tests on notched bars, it was decided to make up two mild-steel specimens, as shown in Fig. 3, and to stress them just to the point where yielding starts and then to show up the flow lines by the Fry etching method. The bars were $1^1/_2$ in. in diameter in the central section. One bar had about $1^1/_2$ in. of V threads, 8 threads per in., while the other had a single notch of the same dimensions as the V threads. These tests should show whether or not the flow in a deeply notched bar of mild steel will be confined to the plane at the bottom of the notch where fracture will eventually occur or to what extent it penetrates into the heavier sections of the bar.

Testing Equipment

Some years ago at the Westinghouse Research Laboratories on the suggestion of A. Nádai special equipment was designed for running constant-strain-rate tests either in tension or torsion or in a combination of both.[3] The machine is illustrated in Fig. 4. The tensile load was measured on the regular Amsler dial while the torsion load was determined by measuring the deflection of two springs attached to the upper head of the machine. An electrical contact device was used on the dial of the Amsler

[3] For this purpose a 10-ton Amsler testing machine was reconstructed by W. O. Richmond and B. Cametti.

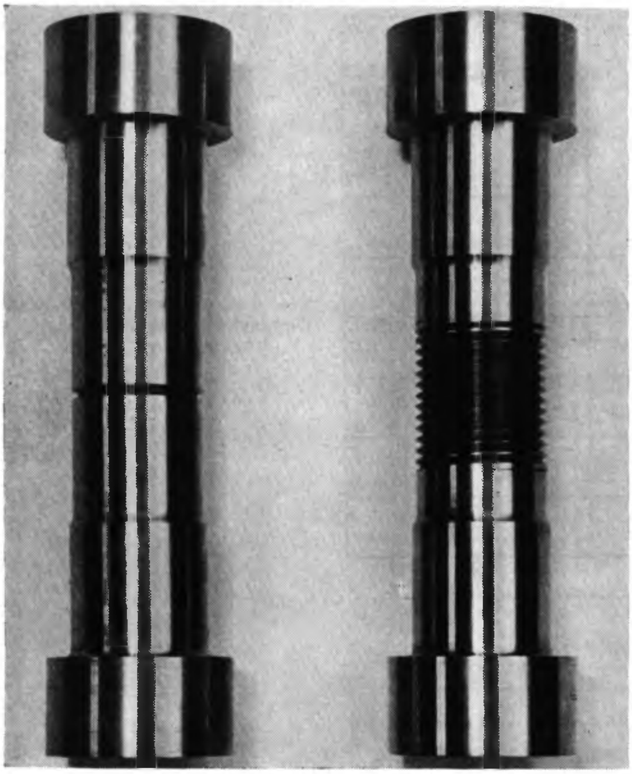


FIG. 3 TEST SPECIMENS FOR SHOWING WHERE INITIAL YIELDING WILL OCCUR

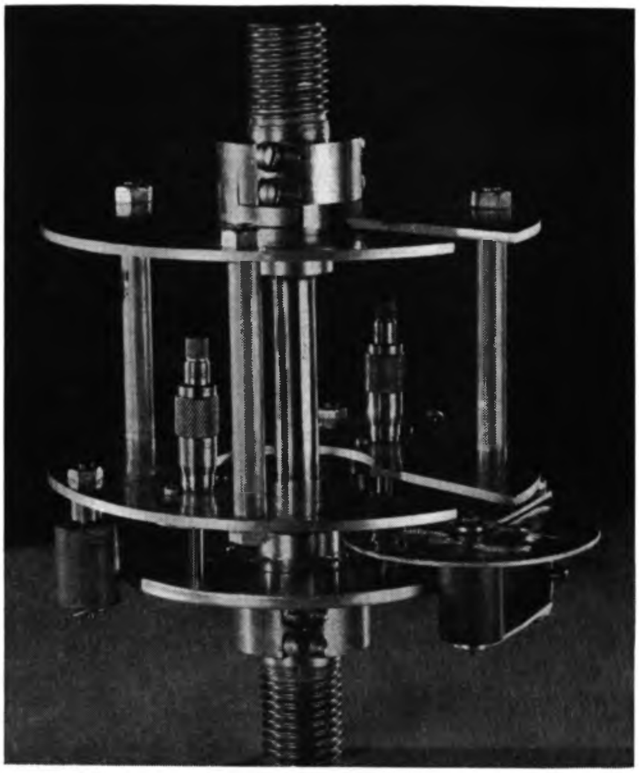


FIG. 5 TENSION-TORSION EXTENSOMETER

load indicator to maintain a constant tension when desired. A similar contact device was arranged on the torsion springs so that a constant torque could be obtained.

The strains were measured by means of the specially designed extensometer shown in Fig. 5. As has already been mentioned, the extensometer was clamped on the shoulders of the test specimen. The extensometer consisted chiefly of a frame which was clamped to the upper shoulder and a plane disk which was clamped to the lower shoulder. The axial strain was determined by two micrometer heads clamped to the frame in such a manner that they could measure the separation between the frame and the plane disk. The angle of twist was measured by a roller attached to the frame which rolled on the periphery of the disk.

TEST RESULTS

Before any combined stress tests were run, the material was tested in pure tension and pure torsion. The pure tension test is curve J in Fig. 6. The ultimate strength was 83,000 psi and the elongation at rupture was 27.3 per cent. In the pure torsion test, curve G in Fig. 7, the breaking torque was 237 lb-ft and the maximum angle of twist was 584 deg.

In addition to the pure tension test, Fig. 6 shows the constant axial strain-rate curves for bars upon which constant torques of 60, 118, and 199 lb-ft were superimposed. The value of the axial strain and the angle of twist at rupture are listed at the end of each curve. It can be seen readily that the amount of axial strain is considerably reduced when a high torque is applied to the bar. The ultimate tensile strength is also reduced. A torque of 60 lb-ft (one fourth of the breaking torque in pure torsion) reduced the ultimate strength only about 5 per cent, but a torque of 199 lb-ft (five sixths of the breaking torque) reduced the ultimate strength over 40 per cent.

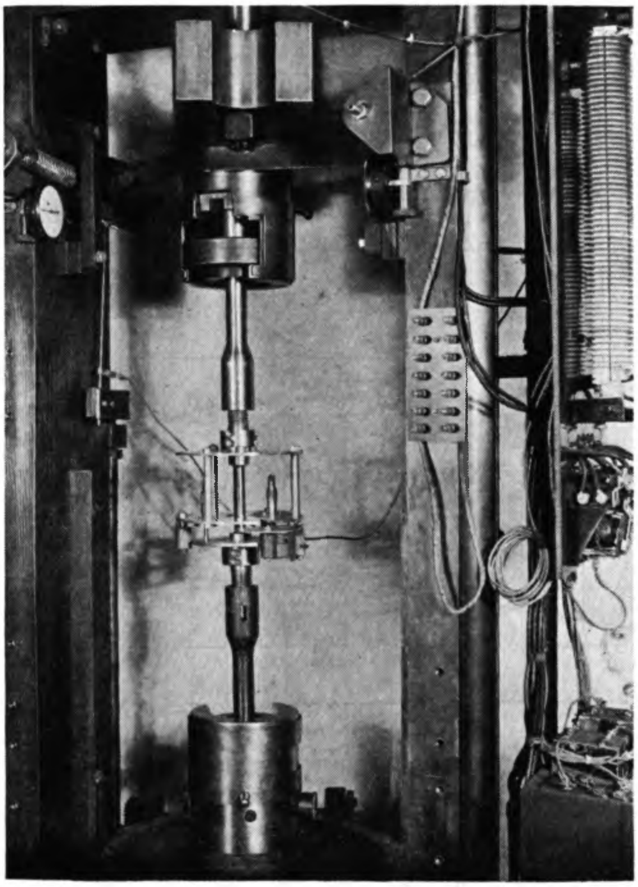


FIG. 4 COMBINED TENSION-TORSION MACHINE

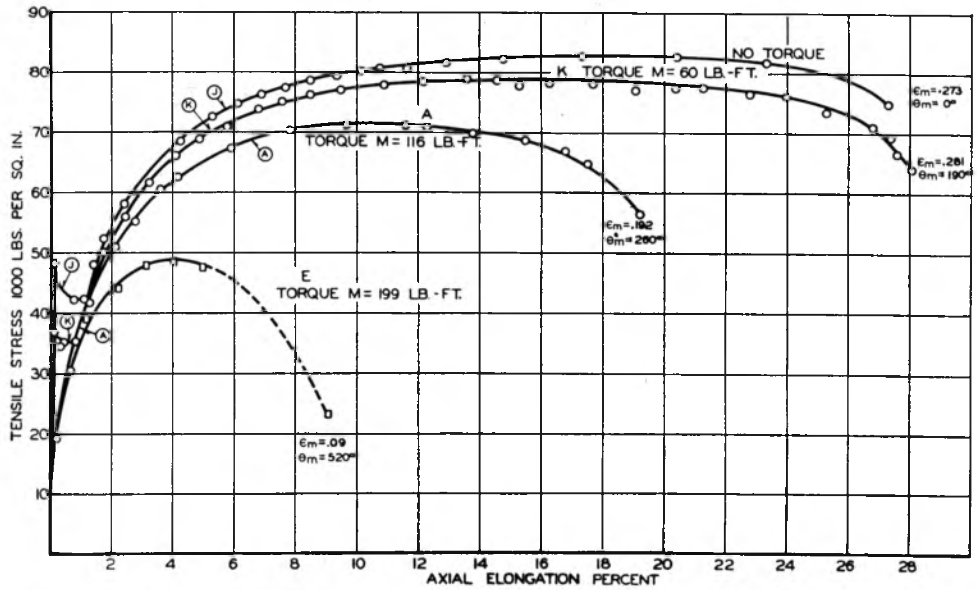


Fig. 6 Combined Stress Tests; Constant Torque Plus Constant Rate Tension

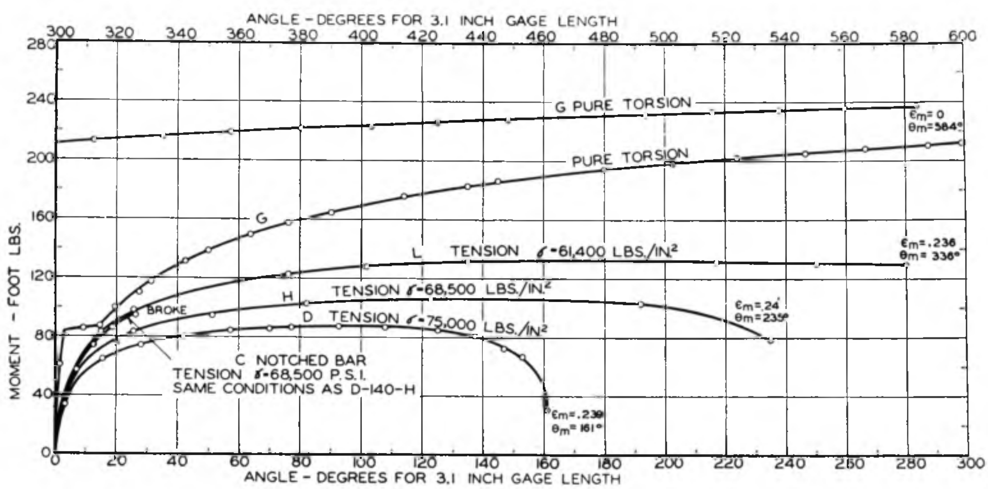


Fig. 7 Combined Stress Tests; Constant Tension Plus Constant Rate Torsion

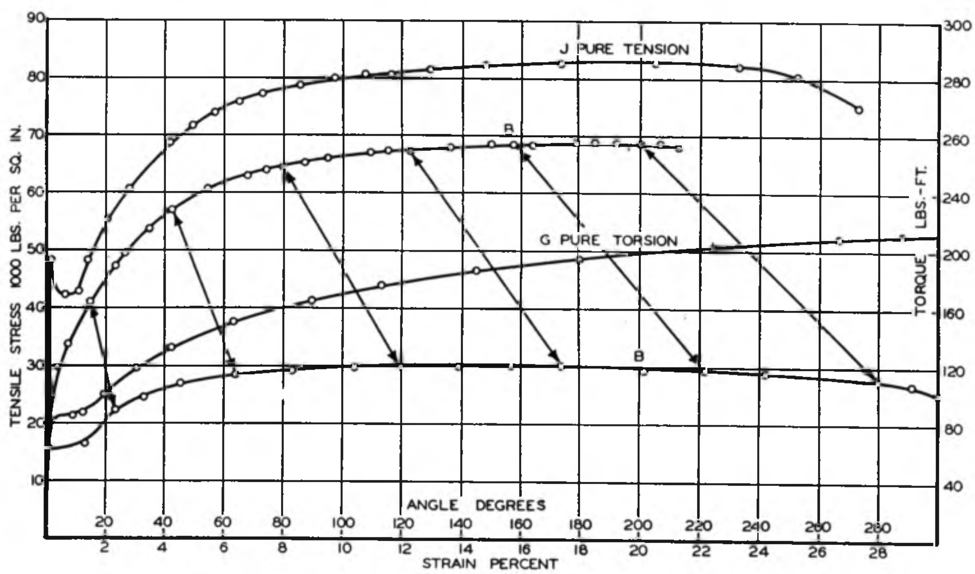


Fig. 8 Combined Stress Tests; Constant Rate Tension Plus Constant Rate Torsion

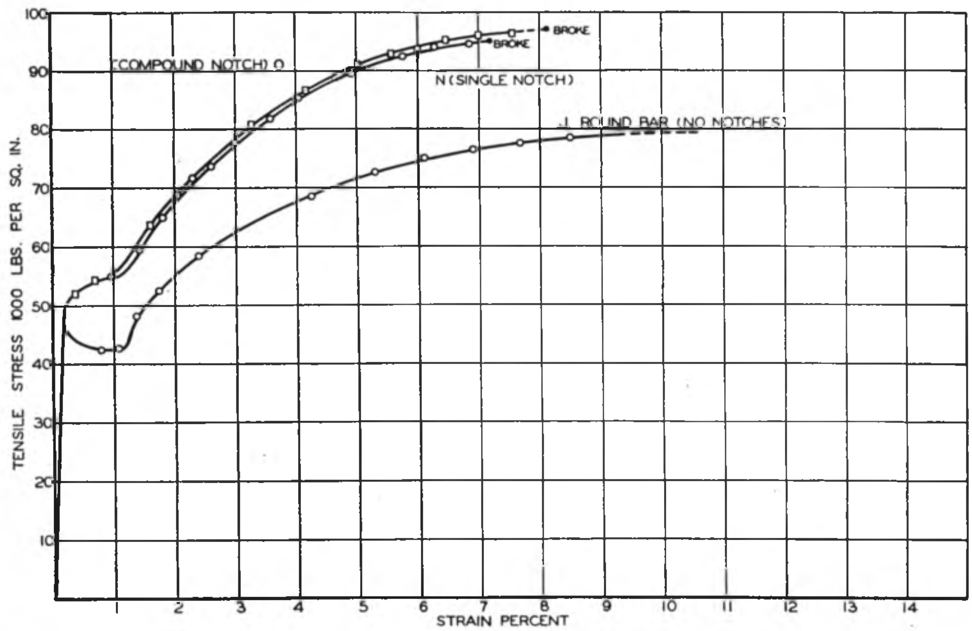


FIG. 9 TENSION TEST ON NOTCHED BARS

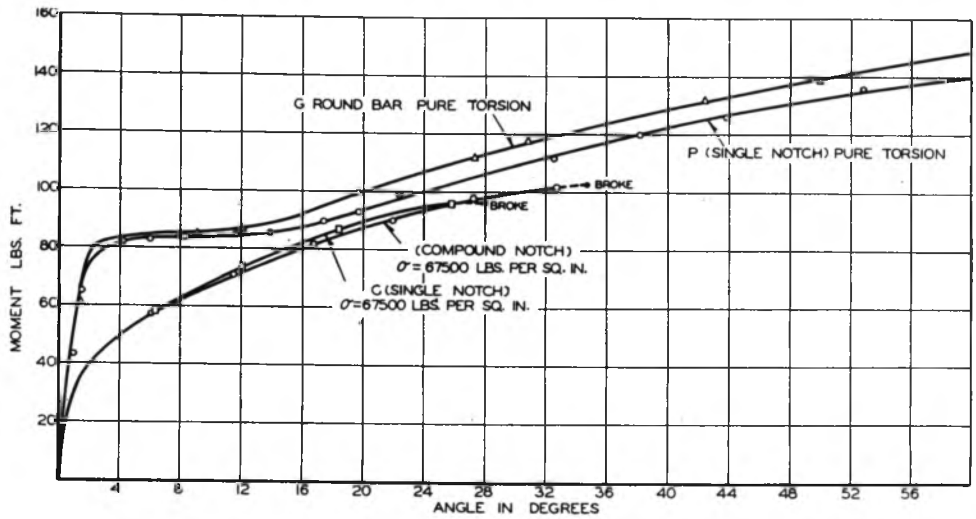


FIG. 10 TORSION TESTS ON NOTCHED BARS

Fig. 7 shows the results of the tests which were run with a constant strain rate in torsion. Curve *G* is the pure torsion test. Curves *L*, *H*, and *D* are torsion tests with superimposed tensile stresses of 61,400, 68,500, and 75,000 psi, respectively. In these tests the tensile loads were fairly high. All were above the yield point in pure tension so that the curves do not show any discontinuity in torsion, such as curve *G* shows. The amount of axial strain did not vary much in these three combined stress tests but the angle of twist at rupture was reduced from 584 deg for the pure torsion test to 161 deg for the bar which also carried a tensile stress of 75,000 psi. Curve *C* shows the results of a test similar to curve *H*. Bar *C* had a single circumferential notch $^1/_{32}$ in. deep at the middle of the gage length. The constant tensile stress based on the area at the bottom of the notch was 68,500 psi. The apparent increase in strength over curve *H* is due partly to the fact that the notch prevents the bar from reducing in diameter and partly to the fact that the actual unit strains in the notch are much greater than shown by the diagram which is based on a 3.1-in. gage length.

In addition to the combined stress tests already mentioned, one test was run with a constant strain rate in both tension and torsion. Curve *B* in Fig. 8 shows this test compared with the pure tension and the pure torsion test. In this figure, the tensile stress and strain coordinates refer to the two upper curves, while the torque and angle coordinates refer to the two lower curves. Bar *B* was being tested in tension and torsion at the same time, hence, there are two curves labeled *B*. The arrows connect points on the two curves which occurred at the same time. For example when bar *B* had a tensile stress of 64,000 psi and an axial strain of 8 per cent, it also was subjected to a torque of 120 lb-ft and had twisted 120 deg in a 3.1-in. gage length.

At this point, an interesting observation can be made concerning the combined tension-torsion tests. In a pure torsion test, such as curve *G* in Fig. 7, the moment-angle diagram has a positive slope until the fracture occurs, and the bar carries its maximum torque just the instant before it fails. In the combined stress tests, where the tensile stress was fairly high, the torque reached a maximum quite some time before failure. This can be seen in curves *H* and *D* in Fig. 7 and in curve *B* in Fig. 8. This may possibly be due to the fact that the diameter of the bar becomes considerably reduced when it is stretched in tension, thus increasing the shear stress without increasing the applied moment.

The results of the tests on notched bars are shown in Figs. 9 and 10. Pure tension tests on the single and the compound notch are compared with that of the cylindrical bar in Fig. 9. Although the bar having the compound notch showed the greatest strength, the difference was but slight. The stress relieving due to the adjoining notches was almost negligible. The notched bars appear to be stronger than the cylindrical bar for the reason already mentioned. Fig. 10 shows the torsion tests on notched bars. Curve *P* is for a single notched bar and can be compared with the pure torsion test (curve *G*). It should be pointed out that these diagrams are moment-angle diagrams and not shear stress-strain diagrams. That explains why the moment curve for the notched bar is lower than the curve for the cylindrical bar. If we should plot the actual stress-strain diagrams, we would probably find that the curve for the notched bar would be higher. Curve *C* from Fig. 7 is replotted here for comparison with curve *M* which is the result of a compound notch tested under the same conditions as the single notched bar *C*. Both had a tensile stress of 67,500 psi superimposed. Here again there is very little difference between the two types of notches.

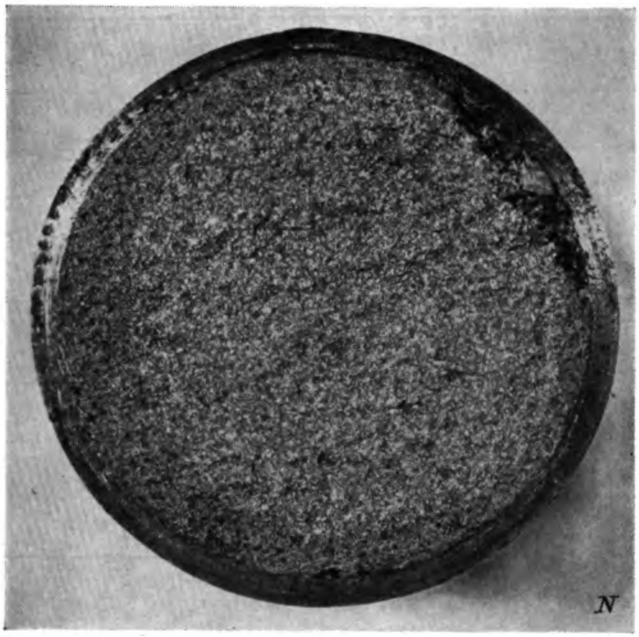
It is quite interesting to note the various types of fracture which occurred in these combined stress tests. Two pure tension fractures are shown in Fig. 11. Bar *J* is the cylindrical bar, while bar *N* had a single circumferential notch. The fracture of bar *J* is a typical tension fracture. In bar *N* two regions in the fracture are discernible. It appears that the bar cracked inward from the root of the notch for some distance and then the bar suddenly pulled apart. The notch prevented any reduction of area and the ruptured surface has the appearance of a brittle fracture.

Bar *G* in Fig. 12 shows how the pure torsion test bar broke. The fracture surface was nearly but not quite perpendicular to the axis of the bar. The upper specimen in Fig. 12 shows the rupture surface of a tension test which had a relatively small torque imposed upon it. This break appears much the same as the pure tension rupture. Fig. 6 shows that its stress-strain diagram was not much lower than the pure tension test.

Two bars which carried a good portion of both tension and torsion loads are shown in Fig. 13. Bar *A* is from a tension test with an added torque, while bar *L* was tested in torsion with an added tensile load. Both broke with a helical fracture; the helix angle, however, was not as great as is encountered in brittle materials in pure torsion.

In Fig. 14, specimen *P* shows the fracture of a notched bar in pure torsion. The rupture surface of bar *M* shown in the lower half of Fig. 14 is probably the most interesting in the group. This was a notched bar tested with combined tension and torsion. The fracture has two distinct regions. Presumably the outer portion where shear stresses predominate failed in shear, while the inner portion which is forced to carry the axial load failed in tension. The phenomenon of the two regions seems to accord with theoretical considerations based upon accepted theories of strength. It is the kind of fracture which might be expected for the case in Fig. 1.

In the tests to show the flow lines in mild steel, the bars were first loaded to the point where yielding had just started. They were then heated to and held at 200 C for 1 hr. Then they were cut along an axial plane through the center of the bar and the exposed sections were polished and etched. The results can be seen in Fig. 15. The portions which have yielded are shown by the dark areas running at about 45 deg to the axis. The dark lines running in the axial direction are probably due to some structure effect or to machining strains and are to be disregarded. In the bar with the single notch, there was some bending stress present and consequently the bar yielded more on one side than on the other. It is evident from this illustration that, for a mild-steel bolt, yielding would start on a conical surface or surfaces. Whether or not a bolt from standard bolting materials at high temperature would start to yield in a similar manner cannot be



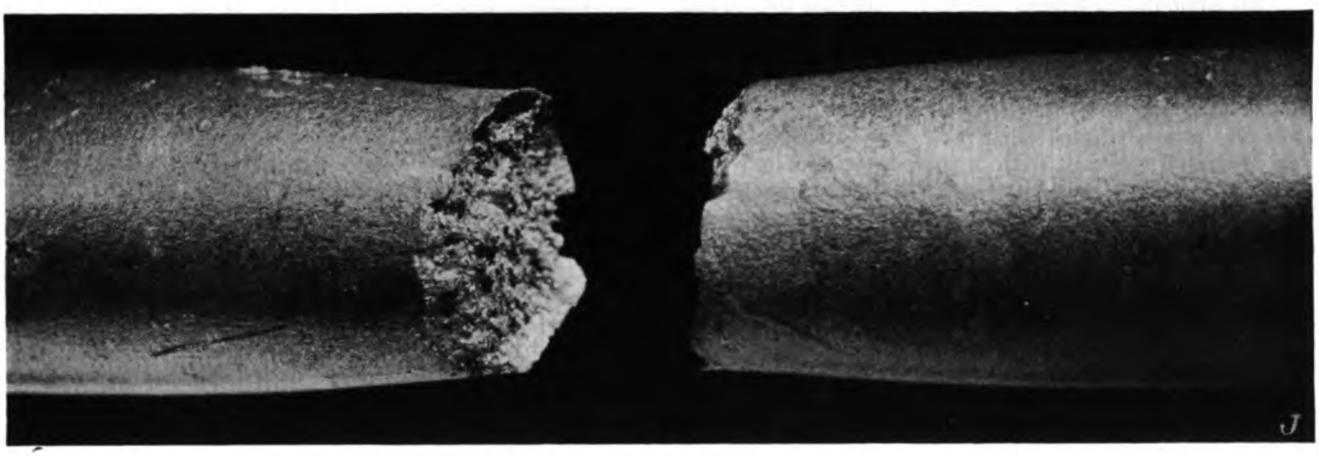


FIG. 11 FRACTURE SURFACES FROM PURE TENSION
(*N* bar with single circumferential notch; *J* cylindrical bar.)

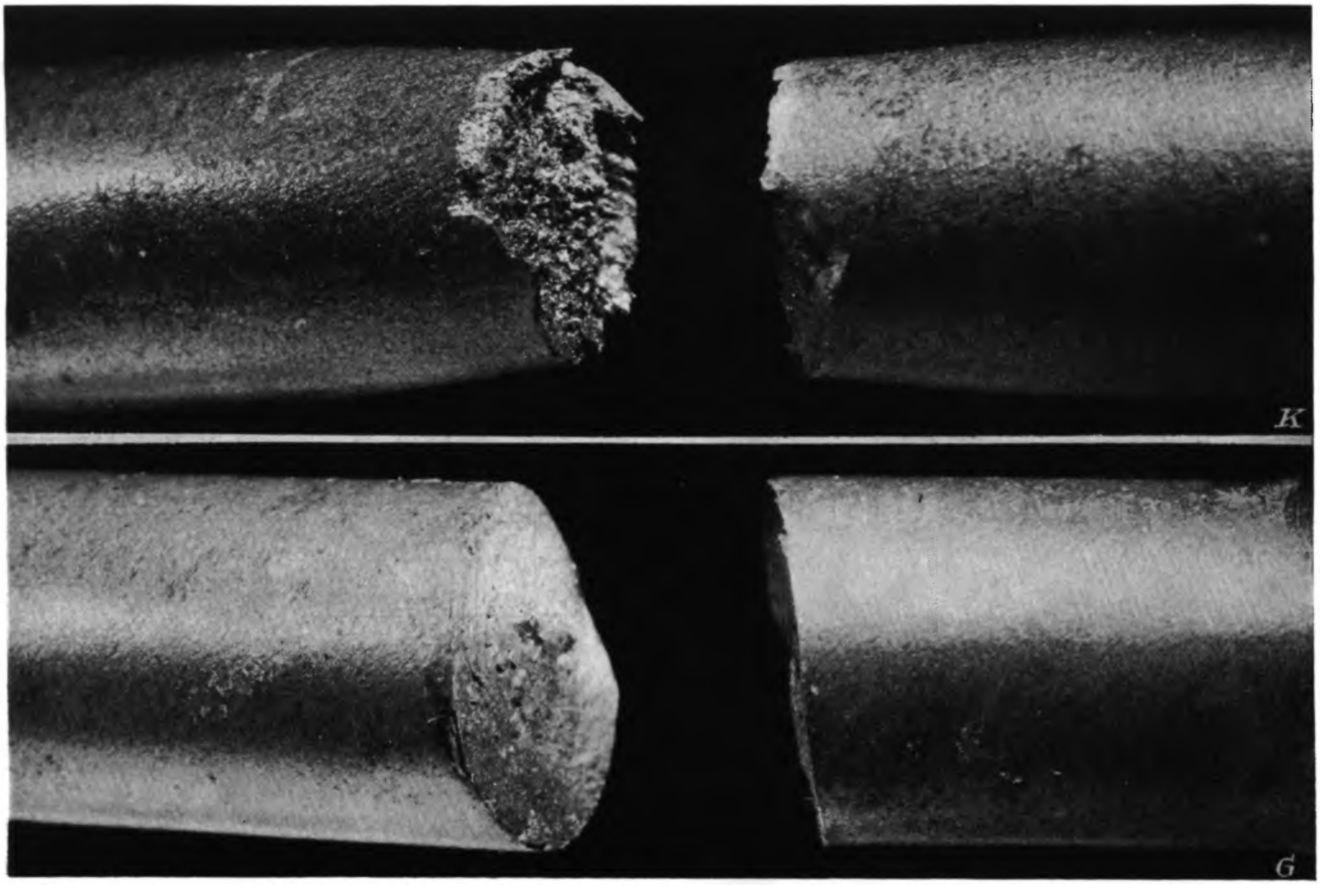


FIG. 12 TYPES OF FRACTURE SURFACES
(Bar *K* shows rupture surface of tension test; bar *G* shows break from pure torsion.)

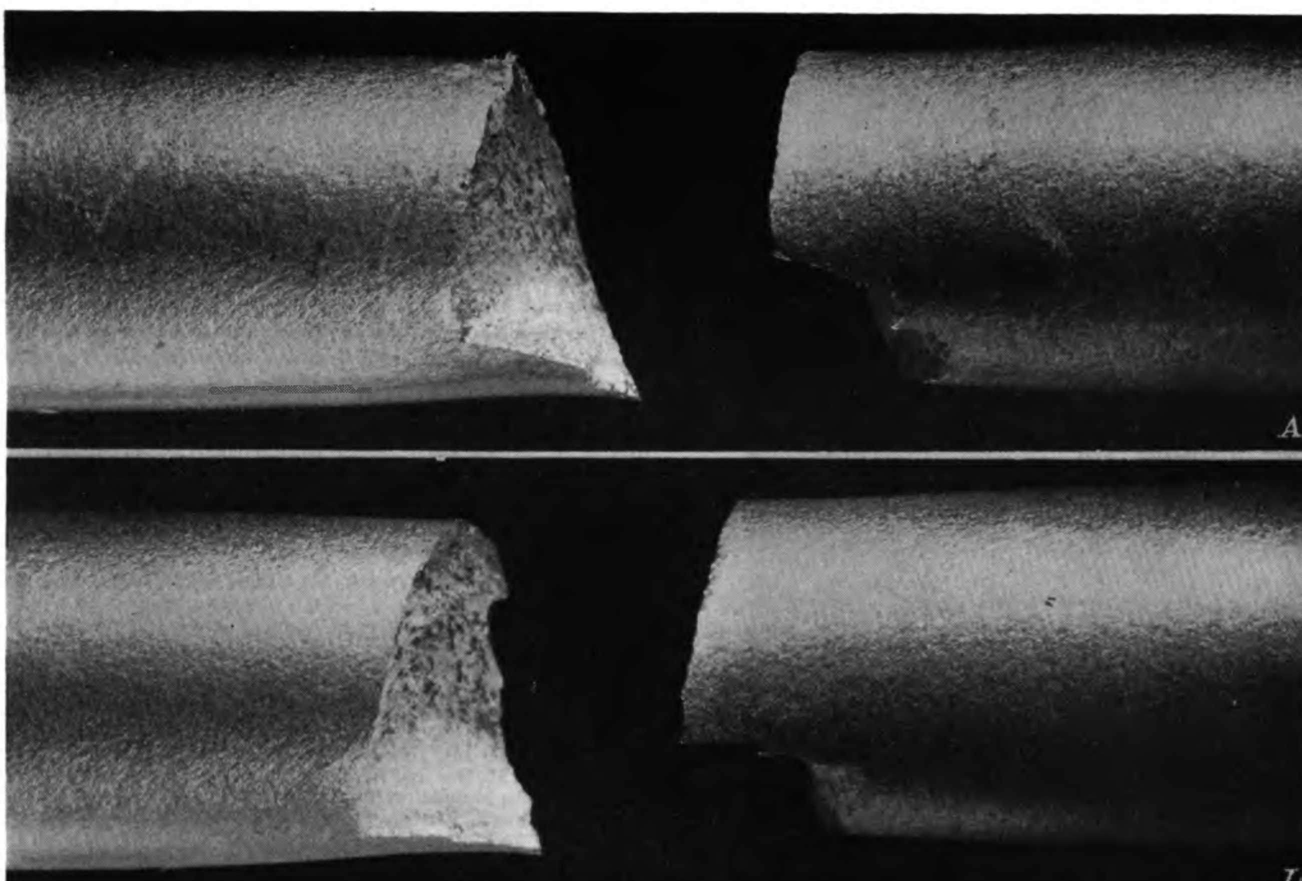


FIG. 13 FRACTURE SURFACES RESULTING FROM TENSION AND TORSION LOADS
(Bar *A* shows rupture from tension test with added torque; bar *L* tested in torsion with added tensile load.)

Fig. 14 Fracture Surfaces of Notched Bars
(Bar P shows fracture of a notched bar in pure torsion; bar M tested with combined tension and torsion.)

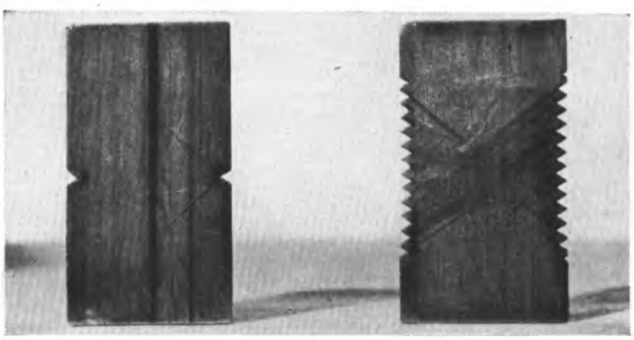


Fig. 15 Initial Flow Lines at Root of Thread

determined, for these slip lines can only be made visible on mild steel which exhibits a marked yield point. These flow lines do not show where ultimate failure will occur, but they point out where the stresses are most unfavorable in the elastic state and where the plastic strains will be localized. Ultimate failure would probably occur along a plane surface perpendicular to the axis of the bar and passing through the notch or one of the threads.

Conclusions

A summary of the test results on the cylindrical bars is shown in Fig. 16. The ratio of the maximum tensile stress in any test to the maximum stress in the pure tension test is plotted as ordinates and a similar ratio of the moments is plotted as abscissas. The test points all fall close to a circle of unity radius passing through the points of the pure tension and pure torsion tests. All points fall outside of a circle the center of which is at the origin and the radius is 0.9.

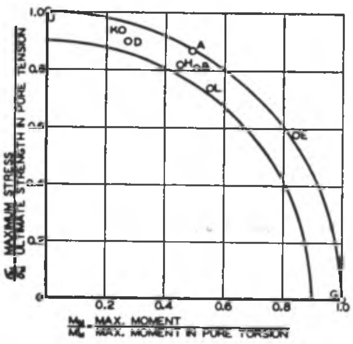


Fig. 16 Test Results

A sharp V notch in a ductile material does not cause any stress concentration that can be detected in a short-time tension test, but it does seriously affect the amount of strain at rupture. This latter statement is particularly true if torsion loads are present. In the pure tension tests, the maximum strain was reduced from 27 per cent for the cylindrical bar to 7 per cent for the notched bar. In the case of pure torsion, the angle of twist was similarly reduced from 584 deg to about 60 deg.

The fact that the stress-concentration factor for the notch turned out to be unity erased any effect which the compound notch might have shown in a more brittle material.

The amount that the bars necked down was roughly proportional to the amount of ultimate tensile load. When heavy torques were present, the bars failed before any appreciable necking occurred. Initial yielding in a circumferentially notched mild-steel bar occurs on a conical surface and not in the plane where the bar will eventually fail.

Acknowledgments

The author wishes to acknowledge the help of Dr. A. Nádai who supervised this work. He also wishes to thank Mr. L. W. Chubb and Mr. F. T. Hague of the Westinghouse Elec. & Mfg. Co. for permission to present these results, and Messrs. G. H. Heiser and M. Manjoine for their assistance in making the tests.

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Discussion

C. W. MacGregor.[4] Referring to Fig. 6 of the paper, stress-strain diagrams are included in which the load divided by the original area is plotted against the over-all strain between the heads of the test specimen for various amounts of applied constant torque. It was found that the total axial strain, as usually defined, is considerably reduced when a high torque is applied to the bar. The writer has been interested recently in plotting stress-strain curves for tensile tests in which the average true axial stress P/A is plotted as a function of the true axial strain ϵ or q' obtained from diameter measurements where $q' = \epsilon = \log \dfrac{A_0}{A}$ (A = actual area; A_0 = original area).

It is realized that the problem of expressing the true stress-strain relationships in the cases treated in the paper is indeed complicated, as mentioned by the author. It would be of interest, however, if the author will include in his closure a comparison of the average true axial breaking stress and strain values as thus defined for the final points in the various curves plotted in Fig. 6. The axial-strain values can be determined from diameter measurements in the constricted portions of the test bars. This may give some additional information as to the effect of torque moment on these quantities.

Joseph Marin.[5] The research reported by the author is particularly important since load-deformation relations for combined stresses are seldom reported beyond the yield point and to rupture. One aspect of the combined-stress problem is the evaluation of working stresses for such a state of stress. In order to do this it is necessary to have some rational theory defining rupture. The author points out that most investigators consider the problem of defining the beginning of yielding rather than final rupture.

Considering the test results reported in the light of theories of failure as applied to rupture, it is of interest to compare tests and theories. In doing this there is encountered the difficulty of evaluating the stresses at rupture. An approximate value of the shearing stress can, however, be obtained by using relations (2), page 578 of the paper for a slope of the torque-twist curve at rupture equal to zero. With this assumption the shear stress is

$$\tau = \frac{3M}{2\pi a^3} \quad [1]$$

where M = torque at rupture and a = radius of specimen.

For tension the stress at rupture is

$$\sigma_x = \frac{P}{\pi a^2} \quad [2]$$

Considering the test results plotted in Fig. 16 of the paper, the equation of the circle representing the test points is

$$(M/M_0)^2 + (P/P_0)^2 = 1 \quad [3]$$

where M_0 = twisting moment in pure torsion at rupture,
and P_0 = axial tension in pure tension at rupture.

Placing values of the stresses from Equations [1] and [2] in [3], the relation between the stresses at rupture is

$$(\tau/\tau_0)^2 + (\sigma_x/\sigma_0)^2 = 1 \quad [4]$$

Using the values of the stresses obtained experimentally, Equation [4] becomes

$$(\sigma_x/\sigma_0)^2 + 1.84(\tau/\sigma_0)^2 = 1 \quad [5]$$

Equation [5] defines the experimental relation between the stress components. The theoretical relation corresponding to this, using the shear theory is

$$(\sigma_x/\sigma_0)^2 + 4(\tau/\sigma_0)^2 = 1 \quad [6]$$

In the same way using the stress theory

$$(\sigma_x/\sigma_0) + (\tau/\sigma_0)^2 = 1 \quad [7]$$

For purposes of comparison Equations [5], [6], and [7], and the test results are shown in Fig. 17 of this discussion. An inspection of this illustration shows that the stress theory agrees best with the test results except for the test in pure torsion. On the other hand, the beginning of yielding for ductile materials is closer to the shear than the stress theory. It is necessary, therefore, to use a different basis in defining working stresses, depending upon whether the yield point or ultimate is considered.

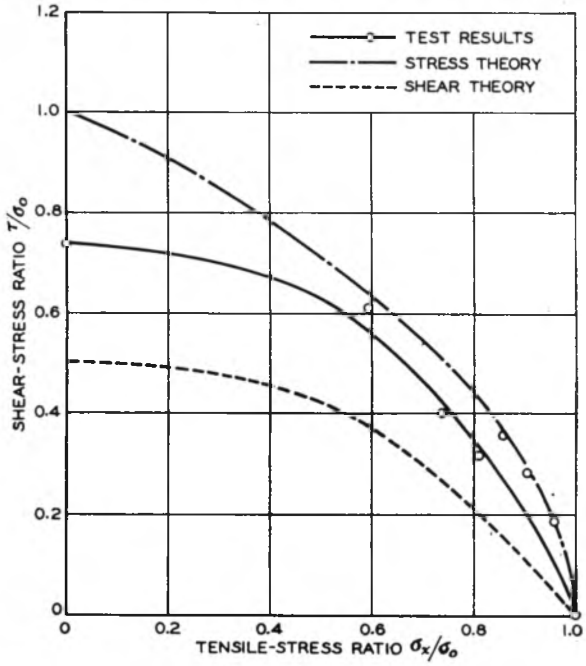

Fig. 17 Comparison of Test Results With Shear and Stress Theories

TABLE 1 ENERGY REQUIRED FOR RUPTURE

Specimen	Tensile-stress ratio σ_x/σ_0	Torque ratio (M/M_0)	Energy in in-lb per cu in. $\div 10^4$
J	1.00	0	2.02
K	0.95	0.25	2.33
D	0.90	0.38	2.12
A	0.85	0.49	2.06
H	0.82	0.44	2.25
L	0.74	0.55	2.37
E	0.59	0.83	3.15
G	0	1.00	3.10

A consideration of the energy required for rupture in the foregoing tests is of interest. Table 1 of this discussion shows the energy in in-lb per cu in. for all of the specimens tested. The values show that there is not a constant value of energy for rupture but an increase from pure tension to pure torsion.

Author's Closure

The writer appreciates the interest shown by Professors MacGregor and Marin in their discussion. The information regarding the dimensions of the necked portions of the plain (those without notches) bars is given in Table 2 of this closure.

[4] Associate Professor of Applied Mechanics, Massachusetts Institute of Technology, Cambridge, Mass. Mem. A.S.M.E.

[5] Associate Professor of Civil Engineering, Armour Institute of Technology, Chicago, Ill. Mem. A.S.M.E.

TABLE 2 STRESS RATIOS BASED ON REDUCED DIAMETERS

Bar	Diam at rupture, in.	Tensile stress at rupture, lb per sq in.	Tensile stress ratio	Shear stress[a] at rupture, lb per sq in.	Shear stress ratio
J	0.425	131,000	1.00	0	0
K	0.425	112,000	0.85	35,800	0.58
A	0.472	79,300	0.60	50,500	0.82
E	0.533	26,000	0.20	64,000	1.04
G	0.563	0	0	61,200	1.00
L	0.463	90,200	0.69	61,300	1.00
H	0.439	112,500	0.86	42,200	0.69
D	0.428	129,000	0.98	18,750	0.31
B	0.465	99,000	0.75	46,800	0.76

[a] Based on Equation 1 of discussion.

In regard to Professor Marin's analysis of the shear stresses at rupture it may be well to repeat again that it is difficult to evaluate the stresses and strains at rupture. If Equation [1] of the discussion is used it must be remembered that in several of the curves the slope $dM/d\theta$ is not zero but is actually negative. If, however, this equation is used in conjunction with the reduced diameter at the necked portion and also if the reduced area is used to compute the tensile stress at rupture the stress ratios given in Table 2 are obtained. These results which are plotted in Fig. 18 will agree much more closely with Equation [6] than with Equation [7] of the discussion. This is evident from the fact that the maximum shear stress in the pure torsion test is roughly one half of the maximum tensile stress in the pure tension test and that most of the points lie close to the circle drawn in the figure. This is not offered as an ultimate analysis of the stresses at rupture but is merely given to point out the improbability that the actual stresses would agree with a maximum-stress theory of strength.

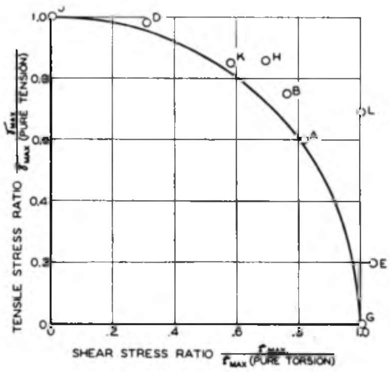

Fig. 18 Rupture Stresses Based on Reduced Diameter

Since the paper was presented, the carbon and manganese content of the steel used in the tests has been determined. Chemical analysis showed that the steel contained 0.47 per cent carbon and 0.71 per cent manganese.

The Engineering Method in Management

By ANDREW I. PETERSON,[1] NEW YORK, N. Y.

Professional management in modern industry demands competence and vital direction of a high order, approached only with scientific objectivity. Engineering and economics are the basic tools of the profession. The author reduces the problems inherent in applied economics to four types: The selection, replacement, and apportionment of elements of production; the analysis of optimum enterprise economy; the rationalization of the dynamics of specific industries; and the interpretation and guidance of the trends and adjustments in the parent social environment. The paper indicates the necessity for more critical research and broader training in industrial management.

THE transformation of a people in primarily local agricultural pursuits to an industrial society with complex and extensive markets is generally designated as industrial revolution. Since engineering has been a principal factor, it has long had concern for the organization, direction, and economy of such enterprise. But the economic problems created by technological progress have taxed our collective ingenuity with recurrent maladjustments sufficient to jeopardize free private industry. Professional management has been described most frequently as industrial engineering, or more elegantly as scientific management. But many engineers unfortunately have concerned themselves with the techniques of their specialties, allowing their responsibilities to management to be those of technicians and clerks. Truly professional management in modern industry demands competence and vital direction of a high order, approached only with scientific objectivity. We must give constant and critical scrutiny to our methods and qualifications.

Scientific method can briefly be regarded as a way of developing the stability of beliefs by minimizing errors and constantly searching for contradictions. We strive for absolute objectivity in a progressive spiral of hypothesis and test. It must be recognized, however, that in the social sciences it is difficult to exclude thoroughly the subjective or normative point of view. Scientific management, by definition directed to the most efficient performance of industrial enterprise, may involve such norms, particularly with its background of extreme individualism, concern for the welfare of labor, and an increasing responsibility for the rationalization of whole industries. Norms of political origin are not entirely separable from concepts of economic organization.

Competition, an Agency of Selection and Development

Competition has been specified as associating struggle with order; an agency of selection and development, if we recognize a Darwinian scheme of life. History describes the mutations of feudal dominance to the ideals of individual liberty to pursue and retain the yields of effort. If individual incentives are supplanted by abstract group responsibilities, it seems realistic to expect that the performance of the efficient would tend to a popular level. We have observed this in our wage experiences, in the recent Russian land experiments, and in many other cases.

That the competition underlying the petty trade of the days of Adam Smith should have suffered distortion and specialization in the evolution of modern enterprise is obvious. It now exists on many fronts of price, quality, advertising, and intangible services. We have groups of individuals in joint enterprise, still motivated individually by hope of gain but under an imperfect development of corporate mechanisms and an insulation of individual risks from control. Institutional presumptions in administration and finance have often driven the other factors of production to positions of marginal disadvantage. We have concentrated jurisdictions over market price with a logical maximizing of profits at a production volume below capacity and at a price above the social optimum. Overinvestment and rigid capitalization are additional concomitants with low utilization of capacity, high unit overhead, and a downward pressure upon price which, once released by a singular act of competition within an industrial group, leads to ruinous price war and instability.

Fundamentally, we appear to have two alternatives, namely, to discard the economic efficiency of large-scale enterprise by attempting a return to the atomistic competition of another century, or better, to adapt our still imperfect enterprise mechanisms for equitable competition in a broader sense than a ban upon "bigness." Many believe that support can be organized for reasonable but more effective police power which, through professional commissions and the use of the consent decree, will permit more efficient solutions of these problems to originate with the experience of business itself. Objective management and professional competence in both private and public practice are vital to the continuance of private enterprise on an economic scale. Failing this, the alternatives are unpleasant to contemplate.

It has taken many years to identify scientific management with a method rather than with a technique. A review of its development and the work of the truly professional men who have carried on the vision of Taylor, Gantt, and Gilbreth would be repetitious. Some of the procedures of recent years, however, seem predicated upon subjective techniques and many of our engineers appear to enter industry with untoward sophistication. The comments in this paper are therefore offered in examination of a few of the basic tools of professional management.

In classifying the infinite range of responsibilities, we find a convergence upon such derivatives as technical engineering, theoretical and applied economics, accounting, statistical analysis, and psychology. Economics and engineering are the warp and weft of the fabric. With more immediate objectives in production, we might visualize the problems of applied economics concerning the engineer as island families of contours, each containing more specific central issues surrounded by successive levels of environment with increasing complexity and scope. In each one, applied analysis is prone to make predictions of representative constants in place of the dependent variables relating them. The problems might be reduced to four types, namely, the selection, replacement, and apportionment of elements of production, the analysis of optimum enterprise economy, the rationalization of the dynamics of specific industries, and the interpretation and guidance of the trends and adjustments in the parent social environment. Such involution demands competent and professional insight rather than pragmatic philosophy.

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Contributed by the Management Division and presented at the Annual Meeting, Philadelphia, Pa., December 4–8, 1939, of The American Society of Mechanical Engineers.

The insularity of specific design problems makes them appear concerned with elementary out-of-pocket economy alone, while in fact highly dependent upon the variables in each circumscribing environment. In selections between new alternatives the simple reckoning of current costs is confused with divergent concepts as to the annual equivalence of depreciable durable assets. Physical lives of general experience are sometimes postulated for probable economic lives in keeping with the obsolescence and actual utility of specific instances. Interest rates upon creditor funds are frequently used in place of anticipated gross rates of return upon capital. Increment- and sunk-cost concepts are imperfectly understood and applied in many situations. In the general employment of minimum-cost-point and break-even-point analyses for multiple alternatives, we have a frequent desire for universal working formulas which lead to mistakes in numerous specific applications. Economic lot-size formulas are frequent offenders in this regard.

Fallacies in Replacement Economy

In replacement economy we appear to have additional fallacies. Each problem strives to compare the significant factors of prospective costs of a new machine, method, or structure with the prospective costs of continuing in service the existing installation. The capital recovery costs in the latter seem properly based upon net realizable value and not the irrelevant book value of accounting, adding of course any modernizing investment necessary to equate reasonably the expected utilities of old and new. The use of book value creates a fiction since the true capital sacrificed by continuance is the realizable value.

There are accessory variations. A few claim that actually recovered capital should be deducted from the first cost of the new alternative in such comparison. This misconception is easily disposed of with the question: How much capital will be actually impounded in the new equipment? Others might profess that these contentions are met by including full book value for the old equipment and, for the new, the old machine's unamortized book value plus first cost. Here the error might arise in equal differences being written off at dissimilar lives in the average case. As to the procedural question concerning the disposition of the discrepancy between book and realizable values, it seems eminently proper to enter it against surplus. The higher book value, born of reluctance to reflect actual current losses in fixed assets, causes overstatement of the operating profit in like degree during the earlier part of the life. Still another form of error lies in the occasional treatment of indirect expense or burden. Being commonly expressed as a rate of some direct cost such as labor, it is often assumed in the case of some labor-saving investment, for example, that an actual reduction in overhead takes place. A realistic approach considers only the absolute changes in each item of indirect expense. The theory and practice of such engineering economy are perhaps nowhere more soundly presented than in the work of E. L. Grant (1).[2] If the degree of rationality which he presents were generally practiced, it would seem that many of the burdensome commitments in plants today would not exist.

In the circumscribing unitary aspects of enterprise we have similar problems, many of them susceptible to advanced quantitative analysis. It is easily established, for example, that the variability of cost functions with output, similar to the natural performance characteristics of machines, is peculiar and stable for every organization and situation. There has been much use of elementary break-even-point analysis, but the apparent lag in extending the introductory method of Rautenstrauch (2), Knoeppel (3), and others is a matter of some curiosity, particularly in view of its success in the management of power-utility industries. Statistical analysis of fixed- and variable-cost performance is vitally significant to finance, budgets, and forecasts. Realism in the internal dynamics of an enterprise is increasingly necessary for flexibility and survival in constantly changing environments.

As our scope proceeds beyond the internal systematics, we embrace the complex ramifications of general applied economics. Even a simple enumeration of the economic and statistical investigations appropriate to modern administration is impossible here. Much of it is of immediate current importance to executives in the timing of purchases, plant investment, or production activity. Other phases relate to the complex significances of change in the general economic environment and the incidence of policies promulgated by enterprises and industries upon the whole economy. We have, however, a variegation of economic doctrine on one hand and on the other a proclamation of half truths and impossible hopes by political extremists. Sincere businessmen are needful of action while scholars procrastinate. It appears inevitable that realistic utility of applied economics must come by way of more advanced economic scholarship in professional management.

Relation of Accounting to Economics

Completely integrated with economic activity is the institution of accounting and its characteristics are pertinent to this discussion. Its principles regard asset values as based upon original cost, except for current assets which are reported at cost or market, whichever is lower. The approach to earning statements is never to anticipate a profit but to provide for all losses. In economics, we fundamentally interpret true earnings or losses as being increases or decreases in actual net wealth. In accounting practice, employing original cost and barring unrealized profit, we have inimical fictions. If assets are reported above their true values, so must be the profits, regardless of the allocation of profits and losses between balance sheet and operating statement. The net worth is out-of-line in either case.

Accountants justify their approaches in various ways. Original costs are often defended upon the grounds of going-concern concepts, undervaluations upon conservatism, and both upon expediency and expense. Going value has had its particular uses but has become a too frequent defense of these discrepancies. In essentially permanent ownerships, a variation in valuation is adjusted by the corresponding long-term level of reported earnings but, with the diverse and short-term ownerships of today, the average stockholder or creditor must rationalize his position with distorted criteria. The claims of conservatism in undervaluation seem equally indefensible upon the same grounds. As for the regular and periodic reappraisal of assets, it has been competently maintained that such procedure need not be impracticable or costly.

Increasing criticism is jeopardizing the future of accounting and the value of much professional engineering research based upon it. It has been stated that accounting must either revise its concepts or be reduced to a level of secondary importance in industry. Some of its spokesmen have attempted to clarify the situation by a sermon of pure conventionalism for the balance sheet and an assignment of predominant accuracy to the operating statement. But there would seem to be an inevitable correlation of their inaccuracies and a futility in such approach. The author is not unmindful of the difficulties illustrated by the conventional requirements of public agencies but the fundamental and necessary realignments are inescapable.

Statistical Method

The third essential, which will be commented upon, is sta-

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

tistical method. We are generally familiar with its importance in all sciences and its powerful modern tools of probability analysis, although some engineering empiricists still popularly consider it as a largely manipulative presentation of data. Its momentous extension to product-quality control in scientific management has evoked a growing realization of its significance in other professional specializations. In time-and-motion studies there are promising speculations of more scientific method. Because of the elementary requisites in equipment and observational skill, and the permitted latitudes of judgment, time-study specialization has grown in popularity with remarkable strides. A good, if not major, proportion of the work done has been viewed as being of questionable value. Managements have in many cases appeared to give insufficient concern to the caliber of young men selected for it. Many of them have been trained in certain techniques of observation but it seems fair to question the number implemented for efficient and objective interpretation. We can easily visualize the potential destruction of managerial control and human values arising from superficial studies.

Time-study observations consist principally of elapsed times for the operating elements of a given job cycle, repeated for reliability from 10 to 50 times. Interpretation aims at establishing a representative time in each standardized element for production planning. The cycle of selected element times is ordinarily adjusted to a good performance level by studies of the concomitant variations of skill and effort. Further transformation produces the long-run performance standard based upon studies of reasonable net productive time per day in each class of work. The final prediction is a basis for wage-rate setting. The variously sponsored techniques are largely empirical and use statistically small samples of observations. It is fully understood of course that the degrees of analysis and accuracy permissible in practice are functions of the size of operations, and that labor's comprehension of the results is a significant factor. The fragmentary speculations raised consider the research that may prove fruitful in establishing more reliable tests and standards of procedure.

Man, like nature, cannot duplicate performance exactly, since many causes operate with unknown frequencies within any one system of conditions. If the latter remains constant, we have some degree of statistical control and establish characteristic limits of variability and credibility to make predictions of repeated performance. Obviously not all systems are found constant or controlled.

In the conventional study of operating elements, we have a row of repeated observations which fluctuate about a central tendency. We face the problem of estimating the underlying universe parameters with reasonable credibility from the corresponding sample statistics, in most cases circumscribed by small-sample errors and some uncertainty as to the form of the distribution function. Care is needed in the selection of sample sizes conforming to the degrees of variation and desired accuracy, and of statistics bearing optimum efficiency in such sample size. The representative time most often used is the mean, sometimes the mode or median.

A few methods attempt to reflect the degree of variation, either to eliminate wild readings or as a part of leveling for continuous performance. The Merrick method, for example, discards extreme readings deviating from their adjacent times by more than approximately 30 per cent of the adjacent times. It also averages the ratios of each mean to its remaining minimum time as a supposedly stable dispersion characteristic of the worker. Superficial tests upon one set of data would appear to suggest that the variation is more peculiar to the operation than the worker. In the description of the Merrick method, there is implied the possibility that minimum times in specific elements are inherently constant for all workers. In view of the increasing fluctuations due to sampling as we go from the median reading to the extreme reading, which are much larger in small samples than in large, it is hazardous to base estimates upon one extreme observation, particularly if arbitrarily selected. If substantiated, this promising hypothesis can be implemented with more reliable statistical tools.

Study of Forms of Time Distribution Needed

The forms of the time distributions also might be more usefully studied. Underlying each operation we have a finite but unknown number of causes contributing time minutiae with individual probabilities of occurrence. The theoretical frequencies of various possible times are those expected in long-run observations under control. Where we have equalized individual cause effects and uniformity in their probabilities of occurrence, the theoretical distribution is based upon the point binomial, positively skewed when the causes are relatively few in number and approaching the bell-shaped symmetrical curve when increased to the limit. In any case it decreases "smoothly" from the mode. When there are predominating causes, these distributions take irregular forms, depending upon the factors present. When we lack a priori knowledge of these causes the distribution smoothness, degree of variability, and symmetry may suggest the degree of control present. Further, the employment of χ^2 against fitted theoretical or graduation functions can give us some test of conformance to control. Do we not have here a promising criterion of the efficiency of motion-study work?

In the establishment of truly representative element times, however, it would seem necessary to make more complete patterns of observation than those taken for short time intervals with one or two operators. The results are dependent upon the time of day, variations from day to day, differences between operators of equal skills and efforts, and actual differences in skill, all in addition to the basic variation within the operation itself. With reasonable care in design, it is possible to take observations over a range of hours, days, operators, and observers by organized sampling, providing complete information for all purposes. Advanced modern methods of statistical analysis presented by R. A. Fisher (4), W. A. Shewhart (5), and others (6, 7)[3] permit practical tests of control, extensive separation of the variables involved, and greater accuracy in determining representative times and variations. Further, the distribution of complete cycle times is a sum function of the element time distributions, where theory establishes their relationships as to mean and variance, depending upon the degrees of correlation between element distributions. With this in mind, the distribution of cycle times has interesting possibilities because of its tendency to greater control. Investigation may support the practical prediction of cycle variance as well as mean time, permitting the use of probability in forecasting performance levels and subsequent operating controls as established in statistical quality-control methods.

It is hoped that these speculations suggest lines of research and a new approach. The statistical competence required for development should not prove unreasonably advanced. As for any misgivings as to the simplicity and economy of such approaches in actual plant applications, the final procedures do not promise any greater involution than other standards resulting from management research.

In general, this paper presents a few thoughts in the direction of more critical research and broader training in industrial management. Engineers in practice, and on college faculties too,

[3] Several working texts for analysis of variance and covariance have recently appeared. The two references cited in the Bibliography are representative.

have a share of human complacence as to their general proficiency in fields outside of immediate experience. Industrial engineering may not have as many of the handicaps of inbred technical specialization, but with three-dimensional responsibilities and opportunities—technology, economics, and scientific management—its success as a profession seems to turn upon a level of fundamental competence which demands the best talent that engineering schools can produce. In management's rapid growth, however, there has been a tendency to conventionalize its practice and research. Rationalization of the rigidities of modern dynamic economy seems to demand a concurrence of experience and scholarship at higher levels than ever before. The marriage of engineering and realistic economics promises a stock that will improve upon the existing parent strains.

The responsibility for development is obviously divided between the engineering schools, industry, and the professional societies. In the first, as recently pointed out by Robert E. Doherty (8), the revision of engineering curricula is hampered by unwarranted insistence upon specialization, vested interest, and rigid viewpoint. To industry would appear to belong the applied specializations in each field, making possible more advanced scholarship in fundamental engineering, economics, and management within the usual span of undergraduate and graduate work. The professional societies bear equal responsibility. It is hoped that the Management Division of The American Society of Mechanical Engineers can continue to support actively a critical and progressive integration of research and publication, interesting the best minds in our profession so that the rest of us may more efficiently serve our common purpose.

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Discussion

E. L. Grant.[4] Two comments occur to the writer as pertinent to the author's observations on incorrect methods in economy studies and his interesting summary of critical views on conventional accounting practices:

1 The voices of those who are critical of the misuse of accounting figures have not yet been heard by a great many of the users of those figures. This statement is based in part upon personal observation and in part upon the confusion of thought evidenced in current writings on such matters as replacement economy.

2 There is no way in which accounting practice can be reformed so that accounting figures may always be used without modification as a safe guide to business decisions. The diversity of business alternatives to be compared is too great for their differences to be measurable by any routine systematic process such as accounting necessarily must be.

A recently observed incident may serve as a simple illustration of the latter point. An engineer redesigned a certain part for a technical product to reduce its cost. The new design effected a $2 reduction in the direct material cost of each unit but involved a $1 increase in its direct labor cost. This he believed to be a $1 cost reduction per unit.

However, the cost-accounting system in the plant allocated indirect manufacturing expense in proportion to direct labor cost; the "burden rate" in this department was 175 per cent of direct labor cost. Thus the $1 increase in direct labor cost was responsible for a $1.75 increase in allocated indirect manufacturing expense, making the net result of the new design an increase of 75 cents in the unit cost as shown by the accounts. Despite this indication by the accounts that the new design was uneconomical, a critical consideration of the facts showed that the company would really save money by using it. None of the individual items, e.g., superintendence, shop transportation, cleanup, heat, light, power, depreciation, taxes, and insurance, which entered into indirect manufacturing expense seemed likely to be altered by the proposed change in design.

At first glance one might be inclined to say that the cost-accounting system should be reformed if it seemed to show costs going up as a result of a policy which actually made them go down. However, careful examination of the circumstances indicated that this was not the correct inference. Considering all of the various purposes of cost accounting, this cost system seemed as good a one as could be devised without excessive clerical expense. What was really necessary was a recognition by management that, to compare specific alternatives, it was essential to look beneath the surface of the cost figures to find the true difference between the alternatives. No possible reform of the cost system could eliminate the necessity of doing this in some cases.

P. T. Norton, Jr.[5] In referring to fallacies in replacement economy, the author correctly states that the capital-recovery costs of the existing installation should be based upon its net realizable value and not upon its irrelevant book value. There are so many ways of proving that the book value of an existing installation should not be used in an economy study, unless it happens that this book value is equal to the net realizable value, that the writer is unable to understand why this obvious fallacy should appear so frequently in discussions of replacement economy.

While the author has proved that any such use of the book value of an existing installation is incorrect, it seems to the writer that there is another way of disposing of this fallacy which should be even more readily understood by the average person. It is obvious that one of the principal purposes in setting a depreciation rate is to bring the book value down to the realizable value at the time the asset is displaced. If the book value is not equal to the realizable value at the time the asset is displaced, it is because the book value is in error by the amount of the difference. Replacement-economy studies should be based upon conditions as they will be if the replacement is actually made and not upon what it was estimated some years ago would be the situation today.

J. A. Piacitelli.[6] In appraising the time-study work, the author points out that a good portion of it has been viewed as being questionable. The writer is in sympathy with this view, for many a study has been rendered incredible by improper methods of observation, inadequate sampling, improper analysis of the

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data, or inappropriate allowances. No matter how much one may do to establish a credible result in any one of these phases, the final answer, or standard, can only be as accurate as the degree of accuracy of each of them. It would be of but slight benefit to work to a high degree of accuracy in the statistical analysis of data when there is doubt as to whether the allowance for fatigue and unavoidable delays should be 10, 15, or 20 per cent.

Viewing the problem as a whole, that is, the development of adequate standards of performance, it is the writer's opinion that attention should be directed toward the standardization of the technique. It appears that we have been too busy standardizing manufacturing operations to get together, examine our own technique, and standardize it. In indicating the desirability of research toward this end, the author has indicated a cause which when removed will contribute greatly toward a more uniform procedure and consistent results.

As for the statistical analysis of data, it is not felt that more accurate methods are needed than the simple ones now at our disposal. Determinations are made for (*a*) the selected minimum, (*b*) lowest third, (*c*) mode or modal average, (*d*) median, (*e*) selected average, and (*f*) the arithmetic average. Which of these is used in the establishment of a standard depends upon the point of view of the analyst. Their use in turn affects the amount of allowance to be applied. For instance, a much higher-percentage allowance must be applied to the selected minimum than the average. What we need is a more common use of the statistical method which would result in the most representative measure of the operator's performance. Investigation by the writer into the relative values of these various statistical methods has led to the adoption of the mode, or modal average, as the simplest means that would yield the most representative results. The modal average, being the value about which most cases recur, is the most reliable method of determining the characteristic or typical performance of an operator. This determination is made possible with relatively few observations as compared with the number that are required when using the method of averaging.

In this connection the writer would like to present for consideration a few facts pointing to the degree of credibility which may be placed upon the modal-average method. What the writer has in mind is the time required to perform an element of a weaving operation in the silk-textile industry. On the basis of slightly over 1000 observations made in approximately 20 silk and rayon mills, the modal average was 0.098 min. Adding to these about 7000 more observations, the modal average was still 0.098, while the arithmetic average dropped from 0.124 to 0.108, or 12.9 per cent. It is interesting to note that the arithmetic average, becoming more representative as the number of observations is increased, approached the modal average to a point within 0.01 min when the quantity of data was increased sevenfold.

Taking at random the observations made on one weaver during one study period, the modal average was 0.116 on the basis of 46 occurrences. When this study was extended to 262 occurrences, the modal average was 0.115 or a change of 0.9 per cent, while the average increased 5.6 per cent. It appears from these data that the modal-average method yields the most consistent results.

The "selected average," determined by the elimination of abnormal readings, is often used as a means of approaching the representative time, but the element of judgment tends to place it on a selective rather than factual basis.

The writer was interested in the author's suggestion pertaining to tests of conformance to control. While it may be desirable to investigate the possibilities of predicting cycle variances "permitting the use of probability in forecasting performance levels and subsequent operating controls," the writer is somewhat pessimistic of the fruitfulness of such an endeavor. A frequency distribution of data is the result of many factors:

1 Degree of standardization of method.
2 Effectiveness of the method (a poor method as well as a good one may be highly standardized).
3 Tools, equipment, and work place.
4 Operator's automaticity in the method.
5 Application or attitude of the operator.

Keeping these factors in mind, the smooth distribution of data, degree of variability, and symmetry may suggest the degree of control present, but it is not believed they will serve as a measure of the efficiency of motion-study work. A symmetrical, smooth frequency distribution with low variability indicates to the writer that the operator is earnestly following a method which is highly standardized. However, it does not suggest that the method is the best which can be devised. Therefore, the writer could not consider the characteristics of the distribution as a criterion of the efficiency of motion-study work. Again, when we have a frequency distribution which is uneven and highly variable, it would appear rather difficult to determine from the array of data whether the operator lacks the required degree of automaticity in the method or is not applying himself, or both.

In closing, may the writer suggest that our immediate attention be directed toward the problem of allowances as one of the steps in the establishment of standards of performance which is far below the recording and analyzing from the point of view of accuracy. A real contribution can be made in supplementing the meager amount of factual data now available to us on this subject. Based as they are largely on judgment and supported by a few scattered actual studies, they tend to nullify the accuracy now possible in our observation and analytical work.

Walter Rautenstrauch.[7] The author quite properly emphasizes the need for a more scientific procedure in dealing with the problems of management. It is true, as he says, that we have been concerned more with techniques of management, that is, formulas and rules than we have with the broader concepts of management out of which may be evolved fundamental and basic principles to guide us in the operations of our industries.

The paper states: "Professional management in modern industry demands competence and vital direction of high order, approached only with scientific objectivity." As a matter of fact, how much scientific objectivity is permitted in the management of business enterprise? Those who are engaged as professional consultants in the field of management are expected to show their clients how to meet the competitive conditions of the market place. These clients, being very "practical" people, expect profits to flow from such advice. Frankly, much of the work which has gone under the name of management, particularly that involving revisions in the methods of wage payment, standardization of jobs, etc., has largely resulted in lowering the wage cost of production and in altering the relative claims of capital and labor to the goods produced to unworkable proportions.

Management engineers have never, except in rare instances, examined into the economic consequences of these so-called systems nor taken into account their effect on the national economy as a whole. The writer fails to see how we can have any scientific objectivity on the larger problems of management so long as management is expected only to produce profits for individual enterprise without regard to the effects on the entire national economy. As the author points out, scientific management should be "directed to the most efficient performance of individual

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enterprises." This raises the question of the units in which efficient performance shall be measured. After an extremely rapid growth in so-called efficient performance of industrial enterprise from 1918 to 1929, we suffered a most disastrous collapse in our industrial economy. Was there real efficiency during this period? What was it we did not take into account as managers of industrial enterprises which caused our industries to break down?

Our engineering societies have been lax in addressing themselves to this most important problem. Perhaps this may be due to a misinterpretation of what the author refers to as the "Darwinian scheme of life." The misinterpretation of Darwin's theory has perhaps done more harm to the national economy than most people realize. The world has yet to learn that only by an intelligent integration of the processes of civilization can we hope to attain a high rate of productivity and an increase in the standard of living. It is true that the solution of the problems of industrial enterprise involves continuous struggle but the object of this struggle, in the writer's opinion, has been falsely directed. It has been directed mainly toward competition between the elements of the national economy, which should be operated on a cooperative basis, rather than upon a competitive basis.

Private enterprise was quick to learn, as it did since the beginning of this century, that its growth and development depended upon an intelligent integration of its departments of operation. How long will it take us to learn that the same intelligent integration of the parts of the national plant as a whole must be accomplished before we can stabilize its performance?

The questions of detail of operation within an individual unit of our economy which the author raises are important and should be inquired into more fully as he suggests. We do need to apply scientific principles to many of the problems that we now solve empirically. We also need to re-examine many of the so-called scientific solutions to determine whether or not they are really scientific.

Very often we have gone into elaborate statistical processes in the solution of problems before we have determined fully the functional relations of the elements about which we are endeavoring to obtain statistical interpretation.

The writer presented a paper[8] before the Philadelphia Section of this Society in 1917, in which was pointed out the imperative need for a closer integration of our banking system with our manufacturing enterprises and the need for their integration on a functional basis. In that paper a quotation was made from the records of The Annals of the American Academy of Political and Social Science, concerning the manner in which the Commonwealth of Massachusetts dealt with the problem of farmer and banker relationship, as follows:

"Several of the banks have opened special farm departments and employed men whose business it is to investigate the applications of farmers for credit. The Plymouth Trust Company, of Brockton, has for two years employed two men, graduates of the Massachusetts Agricultural College, to aid the farmers in applying business methods to the business of farming. The object of the directors of the institution was to get acquainted with them so as to make a businesslike application of credits to those engaged in this important industry. This bank has helped the farmers of their vicinity to buy seed, livestock, etc., and stimulate production by offering prizes to the young people on the farm. It is showing the farmer how to keep cost accounts and how to make cost statements; in short, to know his business both from the technical and from the business standpoint. To worthy persons they stand ready to make a small loan to be used for construction work or for improvement under the supervision of the bank's field agent. Every banker will ask himself, "Does it pay?" It has cost the Plymouth Trust Company $4000 a year net to supply this service to farmers in and about Brockton, but as a result of this and similar activities the deposits have increased in the last five years from $400,000 to $3,000,000."

What forces in our national economy have militated against a further extension of such relationships, not only between the bankers and the farmers but also between the bankers and the manufacturing industries. While it is true that we need better scientific methods to solve the internal problems of industry, we should not close our eyes to the fact that by far the most important problems of management lie in the area of our total economy. While it is almost universally believed that Adam Smith developed the principles of the laissez faire economy, and most people are of the opinion that our economy can be operated most effectively according to these beliefs, it may be instructive to read what Adam Smith actually wrote[9] about some of the problems which he saw developing in the economy of the western world:

> The progress of the enormous debts which at present oppress, and will in the long run probably ruin, all the great nations of Europe, has been pretty uniform.
>
> In the reign of King William, and during a great part of that of Queen Anne, before we had become so familiar as we are now with the practice of perpetual funding, the greater part of the new taxes were imposed but for a short period of time (from four to seven years only), and a great part of the grants of every year consisted in loans upon anticipations of the produce of those taxes. The produce being frequently insufficient for paying within the limited term the principal and interest of the money borrowed, deficiencies arose, to make good which it became necessary to prolong the term.
>
> In consequence of those different acts, the greater part of the taxes which before had been anticipated only for a short term of years, were rendered perpetual as a fund for paying, not the capital, but the interest only, of the money which had been borrowed upon them by different successive anticipations.
>
> A sinking fund though instituted for the payment of old, facilitates very much the contracting of new debts. It is a subsidiary fund always at hand to be mortgaged in aid of any other doubtful fund, upon which money is proposed to be raised in any exigency of the State. Whether the sinking fund of Great Britain has been more frequently applied to the one or to the other of these two purposes, will sufficiently appear by-and-by.
>
> In Great Britain, from the time that we had first recourse to the ruinous expedient of perpetual funding, the reduction of the public debt in time of peace has never borne any proportion to its accumulation in time of war.
>
> When funding, besides, has made a certain progress, the multiplication of taxes which it brings along with it sometimes impairs as much the ability of private people to accumulate even in time of peace as the other system would in time of war. The peace revenue of Great Britain amounts at present to more than ten millions a year. If free and unmortgaged it might be sufficient, with proper management and without contracting a shilling of new debt, to carry on the most vigorous war. The private revenue of the inhabitants of Great Britain is at present as much encumbered in time of peace, their ability to accumulate is as much impaired as it would have been in the time of the most expensive war had the pernicious system of funding never been adopted.

When the debt structure of the nation increases at a greater rate than the rate of production, our economy is in peril and the business of every manufacturer is in jeopardy no matter how scientifically its internal problems are managed. Has there been any scientific inquiry into this problem to which Adam Smith called our attention over two hundred and fifty years ago?

If then we are to have a more scientific approach to the prob-

[8] "Manufacturing in Relation to Banking, Research, and Management," by Walter Rautenstrauch, *Journal* of the Engineers' Club of Philadelphia, Philadelphia, Pa., February, 1918.

[9] "Wealth of Nations," by Adam Smith, 2 volume edition, Oxford University Press, New York, N. Y., 1869. Quotations in order presented are from vol. 2, pp. 581, 583, 585, 587, 594, and 600.

lems of management, should we not inquire into the following types of problems:

(*a*) *A Theory of Organization.* It is unfortunately true that, with all the work which has been done on organization, we have never yet developed a rational theory of the principles of an organized procedure. Until we have a frame of reference or a set of guiding principles, by which to judge the adequacy of an organized procedure in relation to objectives and resources, we can never make much progress in solving the problems of organization.

(*b*) *Operating Characteristics of Our National Economy.* Few attempts have been made to describe how our national industrial economy really functions. Until we understand the functional nature of the sum total of our industrial processes, we will never be in a position to suggest how they may be more efficiently managed or how a particular unit of industry should be operated as a part thereof.

When an engineer examines a particular machine the first thing he does is to find out how it functions, that is, how its parts are related to each other and what are the basic principles underlying its operation. When he understands the functional relationships, he is then in a position, and not before, to derive quantitative relationships concerning its operation.

Management which, up to the present, has been almost wholly concerned with the problems of internal operations is now compelled to give increasing attention to the problem of the individual enterprise as a part of the national economy as a whole if our economy is to be preserved.

When the United States Steel Corporation's net earnings dropped from $195,000,000 in 1929, to a net loss of $59,000,000 in 1932, it certainly was not due to the failure of its management to apply scientific principles to its internal operations. What the managers of the steel corporation and the managers of all other corporations failed to do was to lift their eyes from their own financial statements and observe the march of the economic forces bearing down upon them. After 10 years of disastrous depression, how many managers are familiar with the results of the excellent researches of the National Bureau of Economic Research? How many management and engineering societies have given any consideration to the interpretation of these researches?

The present paper is both timely and important and should urge us to give more careful consideration to the scientific aspects of management problems. It is also high time that we select the most important problems to be solved. Perhaps the Management Division of this Society will help us to select these problems. It is not at all unlikely that, in this search for the important problems of management to which scientific procedure may be applied, we may find the very important question: Management for what?

Author's Closure

To the practical difficulties of recasting general accounting techniques precisely to reflect economic reality, the author readily agrees. But a principal issue remains; that the conventionalized record-keeping so vital to business control, like the shadows of the carried lantern, must be interpreted according to the expediencies of each situation. The oracular pronouncements of its disciples in industrial economics are often in conflict with underlying reality, and erudition as to its detail is sometimes mistaken for professional wisdom. The work carried on by Professors Grant and Norton in the applied economics of engineering selections and replacements and the well-known work of Dr. Rautenstrauch in enterprise characteristics, are encouraging signs. We hope that these are the foundations of a new realistic economics, neither at the extremes of record-keeping technique on the one hand nor abstract "cultural" economics on the other. We need an approach resting upon rigidly scrutinized evidence.

As regards the possibilities of advanced statistical analysis in a more scientific approach to time-study work, the author is interested in the views expressed by Mr. Piacitelli. He indirectly suggests an important issue, in that every investigation, no matter what the precision and objectivity employed in basic interpretation, is overlaid with administrative allowances and the opportunistic motivation of management. That standardization of inevitably personal and subjective factors can usefully be employed is readily granted. That standardization of basic measurement technique can be a guarantee of valid interpretation and homogeneous underlying conditions is respectfully denied. Unassignable or chance causes play a leading role outside of the research laboratory.

Statistical analysis is a vital and powerful tool, not to be casually acquired as a convenient set of arithmetic techniques, nor to be approached with the elegant abstraction of the professional mathematician. Th. von Kármán[10] recently observed: "The engineering application of mathematics is not based on purely logical principles alone; its purpose is the correct interpretation of physical phenomena."

In the scientific aspects of time study, we would appear to aim at an objective determination of the actual sustained basic operating time and the degree of variance of the average good operator under statistically constant underlying conditions. If the latter do not obtain, we can neither combine phenomena in similar investigations nor project them as logical expectations. Since any finite set of observations is merely a sample of an underlying universe of values, we must draw inferences cautiously from efficient statistics which minimize the always present chance variations.

If all of the information underlying the data which Mr. Piacitelli presents were available, it would be expected that a critical appraisal might reveal dissociated conditions which explain the otherwise surprising behavior of the statistics reported. Two possibilities suggest themselves, i.e., that the data have not arisen from the same underlying universe; that the modes have been obtained by the usual approximation techniques, where at best the mode is a very uncertain parameter to estimate. In the first instance cited, the mean of the initial sample of 1000 is given as 0.124 and, by the elementary relationship of the sample means to their weighted mean, we find that the mean of the second sample must have been 0.1057. Also we have the elementary large-sample test of significance for the difference between the means of the two samples under the hypothesis that they come from the same universe

$$T = \frac{\overline{X}_1 - \overline{X}_2}{\sigma_p \sqrt{\dfrac{1}{N_1} + \dfrac{1}{N_2}}} \qquad \text{or} \qquad \sigma_p = \frac{\overline{X}_1 - \overline{X}_2}{T \sqrt{\dfrac{1}{N_1} + \dfrac{1}{N_2}}}$$

where the standard deviation in the denominator is that of the common population. The value T can be expected to be distributed normally and, taking the most favorable maximum value in standard error units, common to tests of significance as 2.576, we would arrive at a standard deviation estimate of 0.2066 which is the minimum permitting an acceptance of the null hypothesis. It would be a very skewed distribution indeed that had only 0.1057/0.2066 or 0.512 standard deviation units between the second sample mean and its zero origin, and only a fraction of this between its mean and the minimum reading or ob-

[10] "Some Remarks on Mathematics From the Engineer's Viewpoint," by Th. von Kármán, *Mechanical Engineering*, April, 1940, pp. 308–310.

servation.[11] Furthermore, we seem to have a uniquely small difference between the mode and mean of this second sample when associated with such a standard deviation. In fact, the employment of the relations between the mean and mode in known functions that are at least nearly appropriate to this type of distribution gives a skewness measure that seems incompatible with the conditions suggested. If the originally estimated standard deviation is actually smaller, the means are significantly different. Such questions pile up to an extent that makes it not inappropriate to raise the issue as to the propriety of considering the data comparable or of combining the two samples of observations referred to.

Mr. Piacitelli's stressing of the importance of establishing standard allowances to supplement the basic operating times is well taken. The author's comments and interest were only in the research possible as regards operating-time measurements themselves and their interpretation, in the use of more critical tests of stability of the phenomena, and in the suggestion of modern analysis. The author is convinced that the latter will not only improve the objective determination of the sustained and controlled good operating levels but permit the isolation of the component variances. Space does not permit attempting an exposition of well-established statistical theory that can be employed but it is to be hoped that students, who are advanced in their grasp of both fields, will find the opportunity at least to investigate another dimension which has been of proved success in other branches of science.

The author does not presume even to a modest competence in the time study that is so tremendously important in our industrial economy and in which Mr. Piacitelli has deservedly attained such eminence. However, if it is to acquire the highly objective statistical tests of research employed in science, a thorough employment of applied statistical theory is necessary. Simple numerical generalization so common to what may be popularly described as business statistics would merely compound the dangers of purely empirical approaches to professional engineering. The author does not recall the savant who remarked that a tragedy is a beautiful generalization wrecked by a single obstreperous fact.

[11] For purposes of simple illustration no distinction is made between population and sample values of standard deviations since the samples are very large.

Fuels for Diesel Engines in Marine Transportation

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After reviewing briefly developments in the field of Diesel fuels over the last 30 years, the authors discuss in detail the present-day requirements for marine-Diesel-engine fuels. A summary of engine troubles and casualties caused by unsuitable fuels is also given. The paper includes a series of tables and explanatory matter concerning Diesel-fuel specifications for various types of marine engines.

THIRTY years ago, hot-bulb-type engines, using solid-injection systems with large-orifice, open-type nozzles, and full Diesel air-injection engines, were operating as pumping engines in the oil fields, using fuels varying from raw crude to 43 gravity kerosene. Many of these engines are still in service. Their efficiencies were relatively low for Diesels, their rate of wear was high, and sometimes they were hard to start, but they burned the fuel at hand and produced the power required with relatively small maintenance and other expense, compared to contemporary steam engines. Like stationary land units, the marine engines of that day were making a name for themselves by burning cheap fuel with an efficiency which has as yet been unequaled except by a few high-pressure high-temperature "modern" steam turbines.

The fuel available at that time was no better and little worse than that available today. Cheap fuel was dirty and contained much residuum. However, cracked fuels had not made their appearance, which normally precluded many present-day combustion and maintenance problems. Clean fuels were all straight-run distillates and gave little trouble, except for injection and combustion difficulties, caused by unsuitably high or low viscosities. Typical fuel specifications for that period, one formulated by Dr. Diesel, are given in Table 1.

During the period when "air injection" was used exclusively, little attention was paid to the properties of the fuel, except that the operating engineer knew from experience the approximate degree of purity and viscosity required to produce passable results in his engines. With the advent of the first commercially successful solid-injection system, manufactured by Vickers in 1914, more attention was given to fuel oil, especially from the viscosity standpoint.

Several investigators published papers covering their work on Diesel fuels during the change-over from air to solid injection about 1918 to 1930, and in 1928 members of this Society formed a committee for the investigation of Diesel fuels with the aim of establishing standard specifications. This committee proposed two fuel-specification standards in 1929, one for light high-speed engines, the second for heavy-duty slow-speed engines. Several series of tests were also made to determine the effect of various impurities on engine operation.

As a result of this and other work on Diesel fuels, a combined A.S.M.E., S.A.E., and A.S.T.M. group submitted several "Diesel-Fuel Classifications," one, in 1934, consisting of five fuel grades, in which ignitibility was considered an important factor for medium- and high-speed fuels. These specifications are given in Table 2.

It was found impossible to obtain sufficient funds to carry out the extensive test program outlined by the joint A.S.M.E.-S.A.E. committee, due to the current financial depression, but, in 1933, a group of men formed the Volunteer Group for C. I. Fuel Research and carried out several series of cooperative experiments during succeeding years on the most promising methods of measuring ignition quality (1).[3] The present ignition quality-test method is described in the 1938 transactions of the A.S.T.M.

The original "Volunteer Group" has now become the Automotive Diesel Fuels Division of the Cooperative Fuel Research (C.F.R.) Committee, Society of Automotive Engineers, whose work was broadened in 1939 to include investigations of the effect of fuel volatility, cetane number, viscosity and gravity on full-scale performance, and engine deposits. Other factors such as the effects of sulphur, gums, carbon residue, and impurities on performance, maintenance, and engine wear are to be investigated in the future (2).

MARINE-DIESEL-ENGINE FUEL REQUIREMENTS

The essential requirements for a marine-Diesel-engine fuel are given in Table 3. They are listed according to their order of use rather than their relative importance.

Storage and Handling

A primary consideration of Diesel fuels is that of storage and handling, as the fuel must be safe in storage, it must not deteriorate, and suitable handling methods must be available.

Fire Hazard. Although a Diesel-fuel vapor has a lower autoignition temperature than some of the more volatile hydrocarbons, fire hazard with Diesel fuel is considerably reduced because of its greatly lowered volatility. Tests have proved that Diesel fuel at its flash-point temperature, being run into a tank with sufficient force to produce considerable foam on its surface, will still not burn, only the foam being burned off when ignited by an electric spark. At temperatures below the flash point, only a small portion of the foam on the surface will burn. Therefore, Diesel-fuel fires, caused by electric sparks or open flames in or adjacent to Diesel-fuel storage, are practically nonexistent.

Aging and Deterioration. Tests covering a period of years have shown that, for the distillate fuels investigated, including straight-run and cracked fuels and their blends, no reduction in the more important fuel properties occurred during long-time storage. Experience shows, however, that certain fuels are incompatible, and when mixed will precipitate compounds which affect engine operation by forming sludge, gums, and lacquers.

Corrosive Properties. All Diesel fuels are not chemically neutral, some containing small amounts of acidic materials. Such acids may effect engine wear and corrode injection equip-

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Contributed by the Petroleum Division and presented at a meeting, New York, N. Y., March 5, 1940, of the Metropolitan Section of The American Society of Mechanical Engineers.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

TABLE 1 PROPERTIES AND SPECIFICATIONS OF EARLY DIESEL FUELS

(Used in large slow-speed air-injection engines)

Property	Date of test 1906	1907	1908	1910[a]	1926[b]	1926[c]
Type of fuel	distillate	crude oil	distillate	...	...	...
Specific gravity	0.852	0.878	0.869	...	...	...
A.P.I. gravity at 60 F	...	...	...	...	26.7	30.2
Cloud point, F	...	...	...	47 Max	...	...
Flash point, F	200	...	150	149 Min	175	175
Viscosity, SSU at 100 F	...	...	...	...	98	50
Water, per cent	...	...	None	1.0 Max	...	...
Water and sediment	...	...	...	...	Trace	Trace
Acidity	...	...	None	...	...	...
Carbon residue, per cent	...	...	...	3.0 Max	3.05	0.4
Asphalt, per cent, Italian tar test	...	...	...	...	45	6
Ash	...	...	...	...	None	None
Insolubles in xylol	...	...	...	Trace	...	...
Total sulphur, per cent	...	0.3	0.17	...	1.34	0.4
Total carbon, per cent	...	87	...	...	...	...
Total hydrogen, per cent	...	12.5	...	...	...	...
Total oxygen, per cent	...	0.23	...	...	...	...
60% distillation temperature, F	...	...	...	572 Max	...	...
Heat value, Btu per lb	19500	19272	19521	16000 Min	19125	19400

[a] Dr. Diesel's recommended specifications.
[b] Great Britain sample.
[c] New York sample.

TABLE 2 A.S.M.E.-A.S.T.M. DIESEL-FUEL CLASSIFICATIONS

(1929–1934–1938–1940)

Property	A.S.M.E. 1929 Progress Report		1934					1938			1940 (Tentative)			
	High speed	Low speed	1-D	3-D	4-D	5-D	6-D	1-D	3-D	4-D	1-D	2-D	3-D	4-D
Flash point, cc, min, F	150	150	legal or 115	150	150	150	150	legal or 115	150	150	legal or 100	legal or 110	legal or 110	140
Water and sediment, max, per cent	0.05	1.0	0.05	0.2	0.6	..	..	0.05	0.1	0.6	0.05	0.05	0.1	0.5
Pour point, max, F	..	..	35	35	35	40	40	35	35	35	0	20	35	35
Viscosity, SSU at 100 F, min	45	..	32	32	..	..	..	33	33	..	..	32.6	..	...
Viscosity, centistokes at 100 F, min	..	..	..	..	..	..	..	..	..	..	..	2.0	..	...
Viscosity, SSU at 100 F, max	100	200	50	70	500	..	..	50	70	250	..	45.5	65.0	140.0
Viscosity, centistokes at 100 F max	..	..	..	..	..	..	..	..	..	..	..	6.0	12.0	...
Viscosity, SSF at 122 F, max	..	..	..	..	..	100	300	..	..	..	..	..	..	...
Carbon residue, max, per cent	1.0	5.0	0.2	0.5	3.0	6.0	10.0	0.2	0.5	3.0	..	..	1.0	3.5
Carbon residue, on 10 per cent residue, max, per cent	..	..	..	..	..	..	..	..	..	..	..	0.25	..	...
Ash, max, per cent	0.02	0.08	0.01	0.02	0.04	0.08	0.12	0.02	0.02	0.04	0.01	0.01	0.02	0.05
Sulphur, max, per cent	2.0	3.0	2.0	2.0	2.0	2.0	2.0	1.5	1.5	2.0	0.5	1.0	1.5	2.0
Ignition quality:														
Cetane number, min	..	..	50	40	30	..	..	45	35	30	45	45	35	30
Diesel index number, min	..	..	45	30	20	..	..	45	32	25	..	..	..	...
CCR, max	..	..	8.1	8.8	9.8	..	..	..	..	..	..	..	..	...
90 per cent distillation temperature, F	..	..	..	..	..	..	..	..	..	..	..	675	..	...
Final boiling point, max, F	..	..	..	..	..	..	..	..	..	..	590	..	..	...
Copper-strip corrosion at 122 F	..	..	..	..	..	..	..	..	..	..	Pass	Pass	..	...
Alkali and mineral acid	..	..	..	..	..	..	..	..	..	..	Neutral	Neutral	Neutral	Neutral

TABLE 3 MARINE-DIESEL-ENGINE FUEL REQUIREMENTS

1 Storage and handling
  A Minimum fire hazard
  B Minimum aging–deterioration
  C Minimum separation–settling
  D Minimum corrosive properties
  E Pumping
    a Adequate fluidity without preheating
    b Minimum sediments and waxes

2 High engine performance
  A High efficiency
    a Balanced ignition and combustion
    b Combustion without exhaust smoke
  B High ignitibility for cold starting

3 Low maintenance
  A Adequate ignitibility minimizing combustion shock, cracked bearings, and engine wear
  B Minimum lacquer, gum-, and carbon-forming tendencies
  C Minimum lubricating-oil sludging

4 Minimum Wear
  A Adequate lubrication for injection pumps and nozzles
  B Minimum abrasive compounds
  C Minimum corrosive compounds
  D Minimum slow, acid combustion

5 Adequate supply—minimum cost
  A Production from domestic crude oils
  B Refining by commercial processes

ment and storage tanks. Also, increased engine maintenance and casualties may occur through the action of metallic oxides which are not always removed by either centrifuging or filtering.

Pumping. Pumping of the fuel must be considered from several standpoints as follows:

a Pumping under cold-starting conditions, particularly in small-boat service.

b Losses in volumetric efficiency of either transfer pumps or fuel-injection equipment, due to excessively high or low viscosities.

c Preheating of the fuel, either in the storage tank or at the engine.

d Precipitation of sediment and waxes on filters and in transfer lines.

Tests have proved (3) that no difficulty in pumping will normally be encountered with fuels having viscosities ranging from 30 sec SU to 200 sec SU, although engine performance will differ markedly with widely varying viscosity fuels.

Pour Point. Pour point is the principal criterion of the fluidity of the fuel at reduced temperatures, although the cloud point should be considered as the minimum temperature permitting certain pumping. Experimental work (4) shows that an exact indication of the pumpability of fuels intended for low-temperature service requires actual low-temperature pumping tests because of the different behavior of the wax structures and inconsistencies in injection failure in relation to fuel cloud and pour points.

Preheated Fuels. Fuels too heavy to pump at existing temperatures may be handled by first heating them, thus reducing their viscosities to obtain sufficient fluidity for efficient pumping. Relatively elaborate heating equipment, however, is necessary, together with high rates of flow. Therefore, heavy fuels are usually used only in large installations. Somewhat reduced maneuverability, due to inability to get under way quickly, increased fuel-bunker and engine maintenance, and danger of excessive engine wear and casualties, practically eliminates such fuels for modern, high-specific-power engines in the military services.

Filtering and Pumpability. Some fuels are difficult to pump because of precipitation of waxes at normal temperatures, which clog filter bags, strainers, and pipe lines. These ill effects can only be guarded against by using suitable heating equipment or fuels having pour points from 15 to 20 deg below the lowest atmospheric temperatures encountered.

Engine Performance

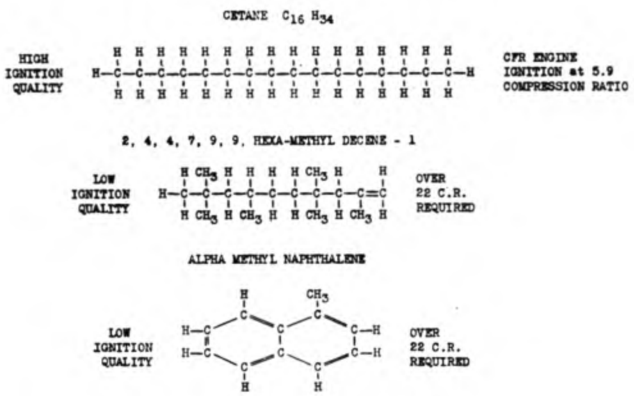
The performance of a Diesel engine is measured by many factors, all interdependent. The effects of various fuel properties on some of these factors have been investigated, while other effects are not yet fully known.

The principal engine-performance characteristics are as follows:

a Specific fuel consumption
b Combustion shock
c Exhaust-gas temperature
d Exhaust-smoke density
e Maximum cylinder pressure
f Cold starting
g Engine wear

The principal fuel properties believed to affect the engine-performance characteristics are as follows:

a Ignition quality
b Viscosity
c Volatility and distillation range
d Carbon residue and impurities



COMPOSITION OF DIESEL FUELS, PERCENT

	Paraffins (C_nH_{2n+2})	Olefins (C_nH_{2n})	Naphthenes (C_nH_{2n})	Aromatics (C_nH_{n-y})
Penna.	70—75	2—5	16—19	4—9
Mid. Con.	50—74	2—14	15—24	3—16
West Coast	40—65	2—18	14—20	10—30

Fig. 1 Diesel-Fuel Chemical Structure and Ignition Quality

Ignition Quality. Ignition quality can be briefly defined as that property of a Diesel fuel which causes it to ignite readily after injection. It is measured by determining the percentage of cetane in a blend of cetane and alpha-methylnaphthalene which produces the same time delay between the beginning of injection and ignition as the fuel under test, a standard C.F.R. Diesel engine being used. An exhaustive treatise on this subject by T. B. Hetzel (5) together with an excellent bibliography has been published.

The factors which govern ignition quality are chiefly chemical:

a Structure of molecule
b Oxidation stability of molecule
c Size of molecule.

Fig. 1 shows the chemical structure of two hydrocarbons, each composed of 16 carbon atoms, one of which has two more hydrogen atoms than the second. The straight-chain saturated compound, cetane $C_{16}H_{34}$, has greater thermal instability than "2,4,4,7,9,9-hexamethyl decene-1," $C_{16}H_{32}$, which is more compact due to the six side chains in its structure. The third molecule, alpha-methylnaphthalene, $C_{11}H_{10}$, is even more compact than the second and, due to its ring structures, will decompose and combine with oxygen only at very high temperatures.

Compounds such as organic nitrates or peroxides are still more unstable in the engine cylinder than cetane, due to free oxygen liberated at comparatively low temperatures. The addition of such compounds in proportions of 1 to 2 per cent will increase the cetane number of an average Diesel fuel by as much as 10 to 30 cetane numbers.

The chemical factors influencing ignition quality cause fuels manufactured from certain crudes to have higher ignition qualities than others. Fig. 1 shows the relative percentages of the four general hydrocarbon families in the three general crude-oil types, which thus provide indications of their true average ignition qualities. The general order of the hydrocarbon families with respect to ignition quality is as follows:

Paraffins—highest ignition quality
Olefins—second highest ignition quality
Naphthenes—third highest ignition quality
Aromatics—lowest ignition quality

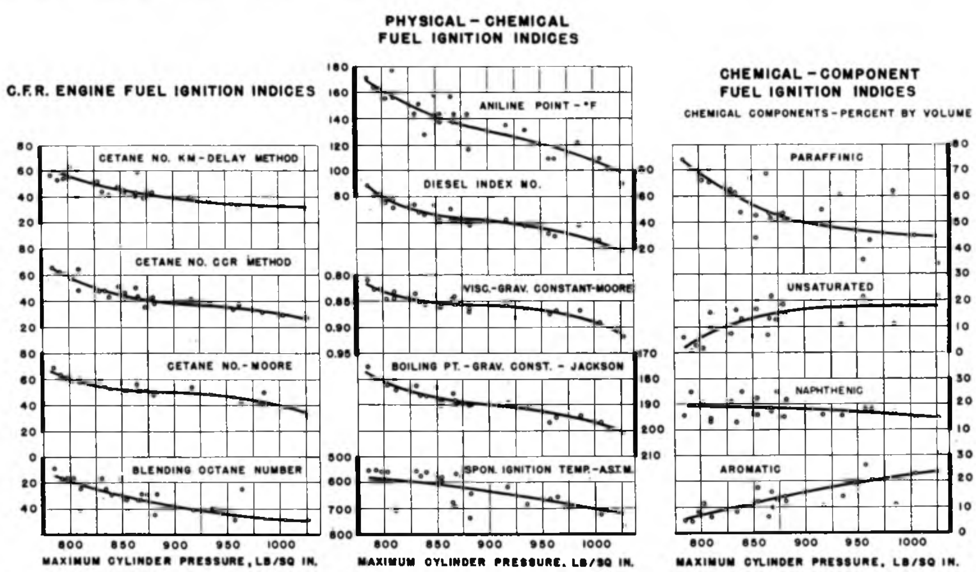


Fig. 2 Correlation of C.F.R. Engine Physical-Chemical and Chemical-Component Fuel-Ignition Indexes of 25 Diesel Fuels With the Maximum Cylinder Pressures Developed in a 4-Cycle Diesel Engine

(Bore and stroke, 4¼ in. by 6 in.; speed, 1200 rpm; 10-bhp single-cylinder engine. Data are for 8 bhp at 1200 rpm.)

Fig. 2 shows the correlation of thirteen methods of measuring ignition quality and the maximum cylinder pressures developed. Inspection of these curves shows that certain of these methods have greater sensitivity per unit of cylinder pressure, and that their data points lie closer to their average curves. They are, therefore, more accurate than the other ignition-quality indexes.

These ignition-quality data were also similarly plotted against the ignition delay of the engine and computed combustion shock.[4] This work indicated that the calculated order of merit of the best seven of these ignition-quality-rating methods was as follows:

1 Boiling point-gravity constant
2 Cetane number, CCR
3 Diesel index number
4 Cetane number (Moore)
5 Viscosity-gravity constant
6 Aniline point
7 Cetane number, KM delay

Since aniline point is included in the now proved, more accurate Diesel index number, and cetane number (Moore) has been superseded by engine tests using cetane, they were dropped from further consideration.

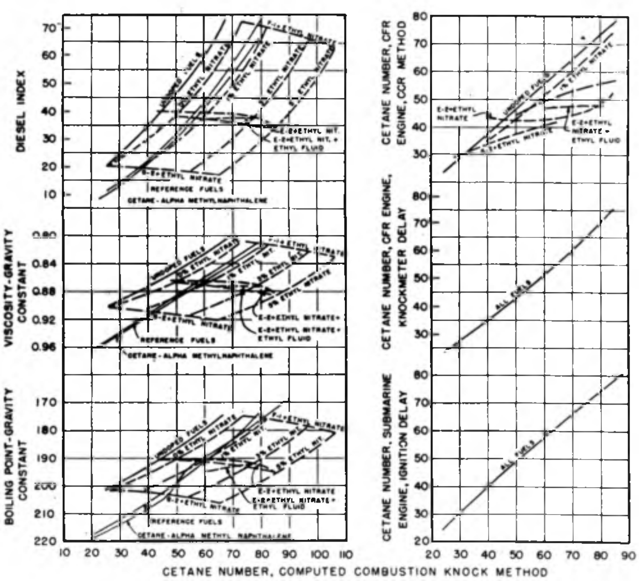


FIG. 3 CORRELATION OF C.F.R. ENGINE AND PHYSICAL-CHEMICAL FUEL-IGNITION INDEXES OF 83 DOPED AND UNDOPED DIESEL FUELS WITH COMPUTED COMBUSTION-KNOCK CETANE NUMBERS DETERMINED IN A 2-CYCLE DIESEL ENGINE

Further tests using doped fuels, Fig. 3, showed that the physical-chemical indexes, i.e., Diesel index number, viscosity-gravity constant, and boiling point-gravity constant, plus the cetane number by the CCR method, were unreliable in predicting the performance of doped fuels. Thus Diesel-fuel ignition-quality determinations must be made in a running Diesel engine, preferably expressed as that percentage of cetane in a blend of cetane and alpha-methylnaphthalene which performs similarly to the fuel under test as regards ignition, i.e., equal ignition delays, equal critical compression ratios (just producing ignition), or other ignition criteria.

The effect of fuel ignition quality on several engine-performance characteristics for four different engines is shown in Figs. 4, 5, 6, and 7. In every case, the maximum cylinder pressures

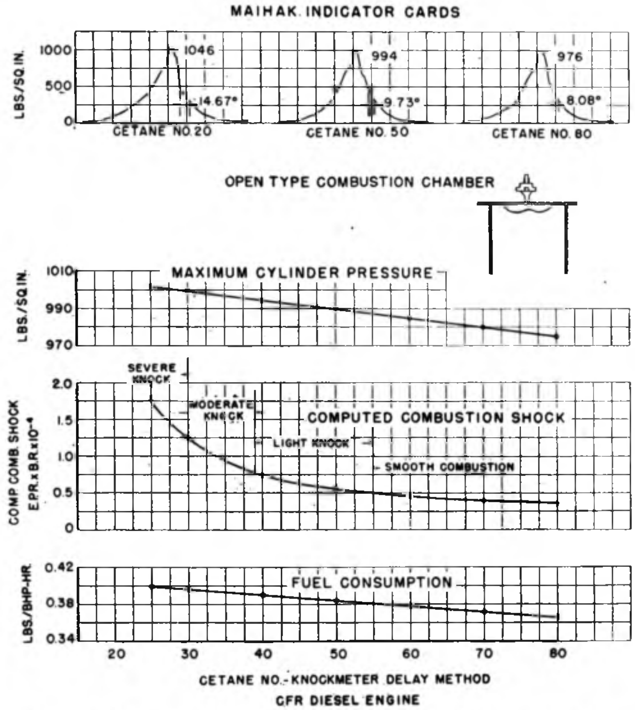


FIG. 4 FUEL PERFORMANCE IN A 2-CYCLE DIESEL ENGINE
(Bore and stroke, 8 in. by 10 in.; speed 750 rpm; 75-bhp one-cylinder engine. Data presented are for 83 fuels, including 24 doped fuels, tested at 75 bhp and 750 rpm.)

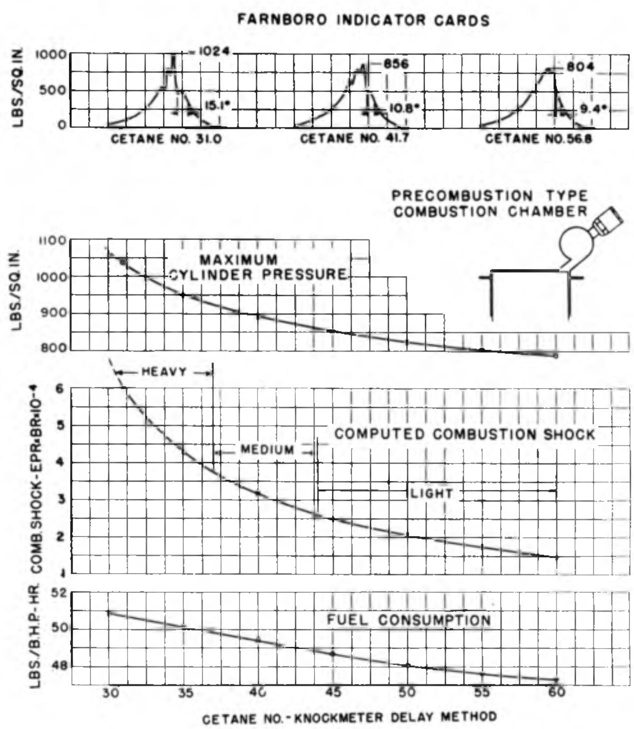


FIG. 5 FUEL PERFORMANCE IN A 4-CYCLE DIESEL ENGINE
(Single-cylinder, $4^1/_4$-in. by 6-in., 1200-rpm, 10-bhp engine. 25 Diesel fuels tested at 1200 rpm and 8 bhp.)

and computed combustion shocks decreased with an increase in fuel-ignition quality. Also, the specific fuel consumption in the open-type combustion-chamber engine, Fig. 4; the precombustion-type chamber engine, Fig. 5; and the air-cell-type chamber engine, Fig. 7, decreased with higher ignition-quality fuels. This

[4] Computed combustion shock

$$= \frac{\text{explosion pressure rise} \times \text{burning rate}}{10^4}$$

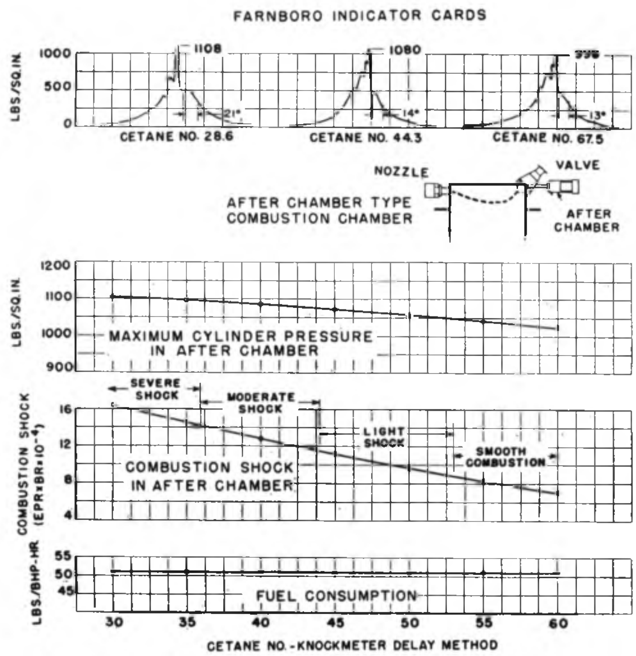


FIG. 6 FUEL PERFORMANCE IN A 4-CYCLE DIESEL ENGINE

(Four-cylinder, 3³/₈-in. by 4¹/₂-in., 1400-rpm, 25-bhp engine. Ten fuels tested at 1200 rpm and 15.74 bhp.)

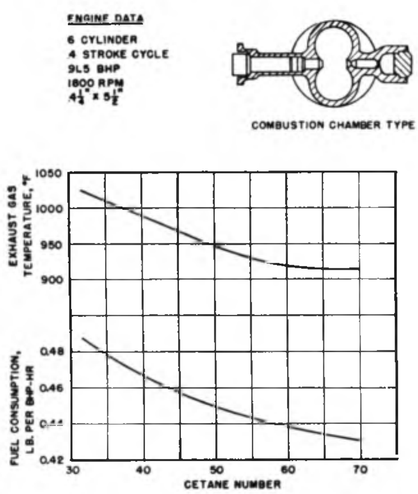


FIG. 7 EFFECT OF CETANE NUMBER ON EXHAUST-GAS TEMPERATURE AND FUEL CONSUMPTION

was not true, however, of an engine using another type of air-cell combustion chamber Fig. 6, as there was no noticeable change in fuel consumption between the limits of 30 and 60 cetane number.

Cold Starting. Fig. 8 shows the results of cold-starting tests on a high-speed marine-Diesel engine. The graph is separated into four regions indicating the relative difficulty of starting the engine. A marked increase in the ease of starting was found as the ignition quality was increased from 20 to about 50 cetane number. Any given cold-starting temperature also required a given minimum cetane number before a start could be obtained. This minimum cetane number decreased as the air temperature was increased.

A wide region of temperatures and cetane numbers was found which produced ignition, but insufficient power due to very late, slow, and irregular combustion to enable the engine to continue running when the starting motor was disengaged. The extreme lower left-hand corner of the graph shows a region where no ignition occurred in any cylinder at any time.

Cetane Number Versus Smoke. Some investigations of Diesel-engine-exhaust smoke (6) show that smoke density is inversely proportional to cetane number, but that other factors have a greater influence on smoke than does ignition quality. This is particularly true of engine design. Fig. 9 shows the effect of fuel ignition quality and engine speed and load on the smoke density of a marine-Diesel engine when using three fuels of 35, 44.4, and 55.4 cetane number which had similar 50 per cent distillation temperatures, i.e., 518, 526, and 525 F, respectively. At full load and speed, 25 bhp and 1400 rpm, the 55.4 cetane-number fuel produced the greatest smoke; the 44.4 cetane-number fuel the next greatest, and the low cetane-number fuel the least smoke. The fuel consumption and exhaust-gas temperature were also minimum at from 30 to 35 cetane number. As the viscosity, carbon residue, sulphur content, and other properties, except for cetane number, were about the same, the difference in smoke density was largely due to ignition quality.

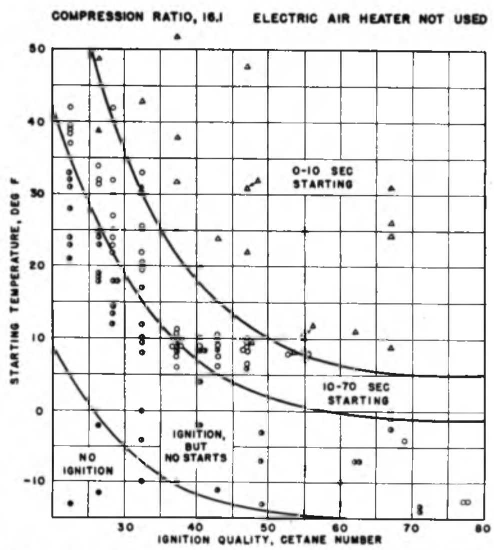


FIG. 8 EFFECT OF FUEL-IGNITION QUALITY ON COLD STARTING OF POWERBOAT DIESEL ENGINE

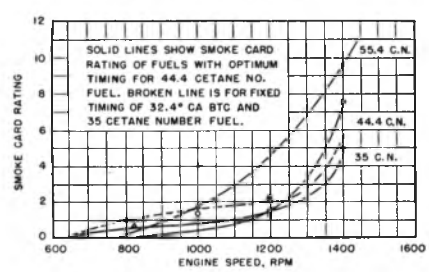


FIG. 9 SMOKE-CARD RATING VERSUS FUEL CETANE NUMBER AND ENGINE SPEED

(Four-cycle powerboat Diesel engine.)

MAINTENANCE AND WEAR

A severe criticism of Diesel engines has been the high weight-to-horsepower ratio required by the high maximum pressure and shock loading imposed on the working parts. These factors also tend to increase maintenance and replacement of parts. It is recognized that proper fuels can alleviate these conditions considerably.

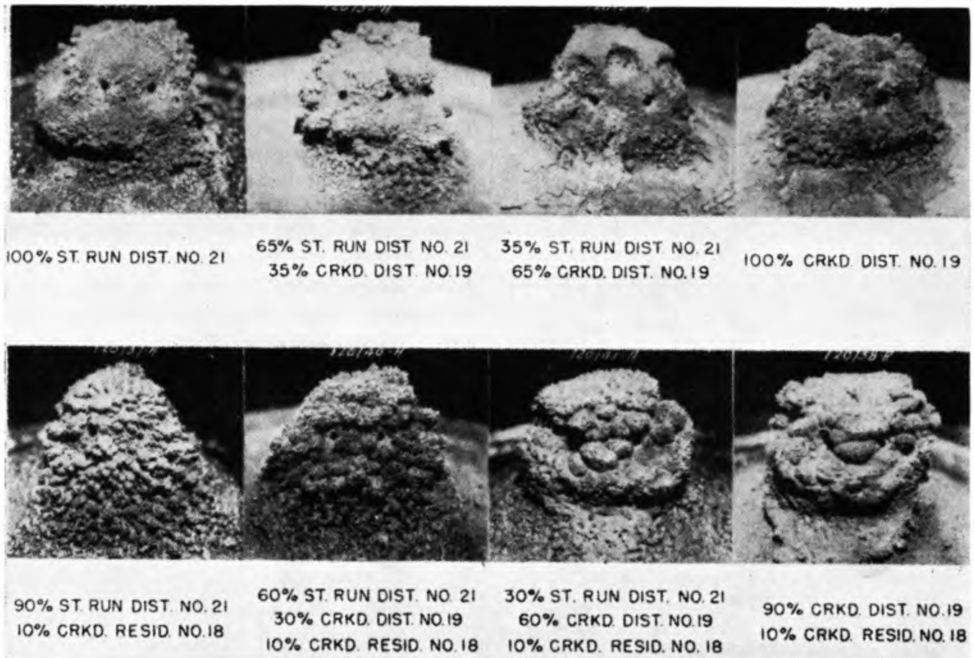


Fig. 10 Fuel-Nozzle Carbonization With Various Fuels and Fuel Blends

Computed Combustion Shock. An index of the effectiveness of the fuel in reducing the maximum pressures and shock-loading has been developed by one of the authors (7), which includes the sudden pressure rise following ignition and the rate of burning during this combustion. This index also correlates well with cetane number, as indicated by Figs. 4, 5, 6, and 7. These show that, in four different engines, an increase in cetane number results in a corresponding decrease in computed combustion

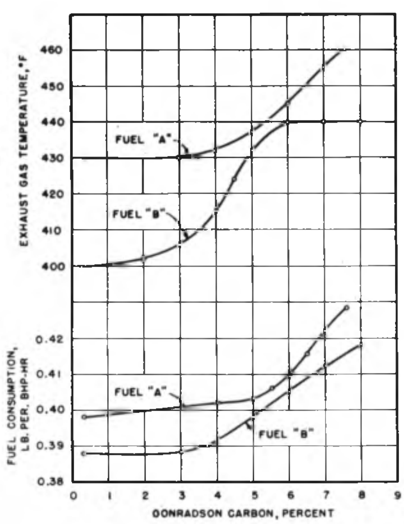


Fig. 11 Effect of Carbon Residue (Conradson) on Fuel Consumption and Exhaust-Gas Temperature
(14-in. by 17-in. airless-injection, 2-cycle, single-cylinder Diesel engine.)

shock and thus a decrease in maintenance, replacement of parts, and a permissible reduction in specific weight.

Nozzle-Tip Carbonization. Fig. 10 shows the wide difference between fuels in forming objectionable carbon on fuel-nozzle tips. Such crater-like formations tend to distort the fuel spray, which in turn produces poor combustion, high fuel consumption, excessive cylinder-head and piston temperatures, reduced power

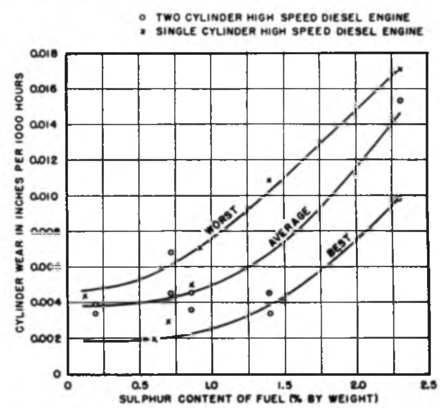


Fig. 12 Effect of Sulphur on Cylinder Wear in High-Speed Engines
(Results drawn from 85-hr tests.)

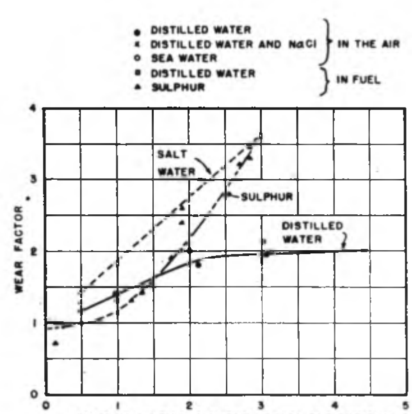


Fig. 13 Influence of Water and Sulphur on Rate of Wear
(One-cylinder low-speed air-injection engine. Wear factor = ratio of wear under conditions investigated to standard conditions when using reference fuel having 0.7 per cent sulphur.)

output, and excessive smoke. These factors all influence engine maintenance adversely.

Carbon Residue. The effects of fuels having high carbon residues (Conradson carbon) are shown in Fig. 11. In a large engine of the type tested, a percentage of carbon residue above about 3 per cent resulted in a decided increase in fuel consumption and exhaust-gas temperature. The true effects of carbon residue are not ascertainable in short-time tests, since carbon residue is closely connected with and involves other fuel properties such as high distillation temperatures, cracked and lacquering hydrocarbon products, and impurities, which cause gradual changes in engine performance and increased maintenance costs and wear.

Sulphur. Figs. 12 and 13 have been selected from the literature (8) and are believed to present some of the best results on the effect of sulphur and water on cylinder wear. It may be noted that sulphur above approximately 0.7 per cent causes a marked increase in the wear rate. These investigators also state that such wear is of a corrosive nature due to the presence of sulphur trioxide in the products of combustion rather than sulphur dioxide as has been commonly believed.

Water and Sediment. The wear of cylinders, pistons, and rings is affected by sand, grit, and dust, either from the fuel, lubricating oil, or the air. An unusual incident of dust affecting marine-engine wear is that of a Mississippi work boat, powered by Diesel engines which experienced excessive cylinder wear; requiring replacements of liners, pistons, and rings at an abnormal rate. After long investigations of fuel, lubricating oil, and other common causes of wear, it was ascertained that the boat was required to pass between long stretches of sandy bluffs overlooking the river. The prevailing winds kept the air full of fine silica which caused the damage. The trouble was then easily corrected by the installation of efficient intake-air filters.

Salt water, whether taken into the engine with the fuel or with the air, accelerates wear considerably as shown by Fig. 13. This action is almost wholly corrosive. Small amounts of chemically neutral water do not appear to affect cylinder wear.

Wear Versus Combustion. The manner in which combustion takes place also affects the wear of Diesel engines. Poor combustion can many times be directly traced to unsuitable fuels, i.e.:

1 Viscosity may affect the deposition of fuel on combustion-chamber walls, since it controls fuel-spray penetration, atomization, and distribution.

2 Volatility controls the rate of vaporization, the richness or leanness of the mixture, and distribution of the fuel spray, and thus the progress and completeness of combustion.

3 Ignition quality controls the timing and location of ignition and, therefore, the condition of the fuel-and-air mixture during the initial stages of combustion. This in turn has a marked effect on all subsequent burning, during which the ignition quality further markedly influences the completeness and efficiency of combustion.

4 High distillation-temperature fractions, cracked products, excessive carbon residues, and some impurities all influence the rate and completeness of combustion.

These major factors determine, from fuel standpoints:

1 The dilution of lubricating-oil films on the cylinder walls and their wear.

2 The deposition of fuel carbon and lacquers on valves, pistons, rings, and cylinder walls, and their sticking and abnormal operation, causing combustion-gas blowby, destruction of lubricating-oil films, increased friction, overloading, overheating, and increased or excessive engine maintenance and wear.

3 The types and amounts of corrosive combustion products formed, causing direct, slight, but cumulative corrosion, during each combustion cycle, on dry cylinder-wall areas and thus their corrosive wear.

Engine Troubles and Casualties Caused by Fuels

As soon as arrangements had been completed for presentation of this paper, letters were written to twenty-six selected companies requesting copies of current Diesel-fuel specifications for marine-Diesel engines and records of engine troubles and casualties caused by unsuitable fuels. While some companies claimed to have had no trouble from fuels, others cooperated by submitting comments and records. Too few records were received to permit of classification by fuel properties. However, Table 4 gives a summary of the essential data derived from this survey:

TABLE 4 SUMMARY OF REPORTS ON ENGINE TROUBLES AND CASUALTIES CAUSED BY FUELS

Engine troubles and casualties		Fuel-property causes
1 Excessive carbonization:		
a Exhaust valves and piston rings	(PC)[a]	High carbon residue; hard asphalt; cracked, lacquering compounds; impurities
b Fuel nozzles	(KC)[b]	High carbon residue; hard asphalt; cracked, lacquering compounds; impurities
2 Lacquer formation:		
a Excessive lacquering of engine in few hours prevented operation	(KC)	Fuel was known cause; probably highly cracked fuel or a blend containing residual products
b Stuck fuel pump plungers, prevented engine operation	(KC)	Fuel was known cause; probably contained lacquering gums
3 Corrosion, pitting:		
a Fuel - pump plungers, valves, and fuel nozzles were badly corroded and pitted	(KC)	Fuel contained saline salts
4 Poor combustion:		
a Lubricating trouble; wear	(KC)	Excessive viscosity; probably also high carbon residue; high distillation end point; impurities
b Bearing failures	(KC)	High distillation end point; low cetane number; high sulphur
5 High maintenance:		
a Fuel-injector valve sticking; cracked cylinder heads	(KC)	Incompletely refined, corrosive fuel; plus small amounts of water and silica
b Generally increased maintenance	(PC)	High carbon residue; hard asphalt; cracked, lacquering compounds; impurities
c Generally increased maintenance	(PC)	High carbon residue; hard asphalt; cracked, lacquering compounds; impurities

[a] (PC) = probable cause.
[b] (KC) = known cause.

Since more is often learned from troubles and casualties, if analyzed, than from "clear sailing," it is hoped that a thorough, cooperative survey and classification of troubles and casualties caused in engines of various types by Diesel fuels may be made and correlated with the unsuitable fuel properties, in a future paper.

DIESEL-FUEL PROPERTIES AND SPECIFICATIONS

Major Classifications

There is considerable evidence to support the belief that Diesel fuels can be separated into four major classes, as follows:

1 Fuels requiring heating and centrifuging or other cleaning before use.

2 Fuels for large, slow-speed engines; about 14 in. cylinder diam or over and speeds below about 300 rpm.

3 Fuels for medium-size, medium-speed engines; about 6 to 14 in. cylinder diam and 300 to 1200 rpm.

4 Fuels for small, high-speed engines; below about 6 in. cylinder diam and speeds from about 1200 to around 2600 rpm.

Low cost is the principal reason for that class of fuels requiring heating and cleaning. Such fuels are not usually suitable for Diesel engines without thorough settling, centrifuging, or filtering and heating by the user. There are sound technical as well as economic reasons for the existence of the slow-, medium-, and high-speed-fuel classes:

1 Differences in the time available between various engines for all injection, ignition, and combustion processes.

2 Differences in cylinder sizes and types of combustion chambers.

3 Differences in the tolerations of kinds and amounts of deposits formed on pistons and cylinders and in maintenance and wear.

Time Available. A basic difference between all types and sizes of Diesel engines is speed or rpm. In large, slow-speed engines, each crank-angle degree is equal to a longer period of time than in a higher-speed engine and thus produces many effects on engine performance, viz.:

1 Ignition delay, instead of comprising a period of from 15 to 20 deg of crank angle as in the high-speed engine will only require from 2 to 5 deg, thus greatly reducing combustion shock and maximum cylinder pressure.

2 Lower rates of injection are possible which again tend to reduce maximum cylinder pressures and effect better combustion control.

3 Combustion will be completed much earlier in the power stroke, which produces higher thermal efficiencies, lower exhaust-gas temperatures, and cleaner exhaust gases. It further decreases exposure of portions of the dry areas of the cylinders to the corrosive gases formed as intermediate products of combustion.

4 More complete burning of the fuel is possible, tending to eliminate many of the partial products of combustion which accelerate cylinder wear such as carbon, gum, and organic acids.

These effects greatly reduce the importance of narrow and restricted limitations on fuel ignition quality, viscosity, volatility, and carbon residue.

Cylinder Size and Combustion-Chamber Type. Increases in cylinder sizes result in higher compression temperatures due to reduced heat losses on the compression stroke, better mixing of fuel and air, and less impingement of fuel on cold walls. The ultimate results are, again, shorter ignition delay, lower injection rates, and shorter and more complete combustion. Thus, fuel properties are affected by reducing the importance of narrow and restrictive specifications.

Differences in Classifications

Number and Range of Properties Specified. Because the importance of fuel properties increases with engine speed, the number and range of property values change accordingly. Thus, high-speed-fuel properties are controlled by 12 to 16 property specifications and low-speed and heater-type fuels by 3 to 7 specifications. The spread allowed in fuel-property values is usually very great for slow-speed as compared with high-speed fuels. The range in viscosity for a slow-speed unheated fuel is from 40 to 150 sec SU, as compared to a high-speed range of 35 to 45 sec SU.

Amount of Impurities. The maximum allowable amount of impurities may be increased in slow-speed fuels as compared to high-speed fuels and this is generally done as shown in Table 5 which gives average values from several typical specifications in each class.

Cost. A major consideration when choosing a Diesel fuel is over-all cost, which includes the cost of the fuel and the effects of the fuel on maintenance, replacements, and efficiencies. There is little doubt that a slow-speed fuel will be the most economical to use except where lower costs, including maintenance factors, have been obtained from the use of the heater-type fuels.

TABLE 5 AVERAGE VALUES OF IMPURITIES FROM TYPICAL SPECIFICATIONS

Property	Fuel Type: Cleaning required	Slow-speed engine	Medium-speed engine	High-speed engine
Sulphur, per cent	2.4	2.0	1.5	0.75
Water and sediment, per cent	0.5	0.5	0.3	0.1
Ash, per cent	0.2	0.1	0.03	0.01
Carbon residue, per cent	9.0	3.0	1.0	0.1

TABLE 6 PROPERTIES AND SPECIFICATIONS OF DIESEL FUELS REQUIRING HEATING AND CENTRIFUGING OR OTHER CLEANING

Property	Bunker C	Fuel oil No. 6	British B.S.I. fuel oil No. 3	Bunker B
Flash point, F	210	248	150 min	200 max
Viscosity, SSU at 100 F	...	760	840 max	1000 max
Viscosity, SSF at 86 F	160	...	.....	.....
Viscosity, SSF at 122 F	44	39	.....	50 max
Water and sediment, per cent	0.5	...	1.0 max	0.7 max
Specific gravity at 60 F	0.943	1.010	.....	.....
A.P.I. gravity at 60 F	...	...	.....	13 min
Ash, per cent	0.057	0.4	0.1 max	0.05 max
Carbon residue, per cent	6.5	12.9	8.0 max	11.0 max
Sulphur, per cent	1.14	2.39	.....	1.5 max
Heat value, Btu per lb	18563	18275	18250 min	.....
Pour point, F	...	−4	.....	.....
Diesel index number	...	14	.....	.....

TABLE 7 DIESEL-FUEL SPECIFICATIONS FOR LARGE SLOW-SPEED ENGINES

Property	A.S.T.M. 1939	British B.S.I. 1937	Manufacturer 1940	Refiner 1940	User 1940
Flash point, cc, min, F	140	150	150	150	150
Water and sediment, max, per cent	0.5	1.0	0.6	0.5	1.0
Viscosity, SSU at 100 F	140 Max	840 Max	40–150	100 Max	65–130
Carbon residue, max, per cent	2.5	8.0	3.0	1.0	0.75
Ash, max, per cent	0.05	0.10	0.04	0.02	0.1
Pour point, max, F	35	...	35	0	10
Sulphur, max, per cent	2.0	...	2.0	2.5	1.5
Cetane number, min	30	...	35	...	...
Acidity	Neutral	...	Neutral	...	...
Heat value, Btu per lb, min	....	18250	....	...	...
Initial boiling point, min, F	....	...	375	...	...
90 per cent distillation temperature, max, F	....	...	725	...	...
Specific gravity at 60 F, max	....	...	....	0.935	...
A.P.I. gravity, min	....	...	....	19.8	24–28

TABLE 8 DIESEL FUEL SPECIFICATIONS FOR MEDIUM-SPEED, MEDIUM-SIZE ENGINES

Property	A.S.T.M. 1939	British B.S.I. 1937	Foreign navy	Manufacturer	Refiner	User
Cetane number, min	35	...	...	...	....	50
Aniline point, min, F	....	1.13	...	...	....	....
Viscosity, SSU at 100 F	65 Max	69 Max	60 Max	35–75	80 Max	35–45
Carbon residue, max, per cent	0.5	3.0	0.5	1.5	1.8	....
Carbon residue on 10 per cent residue, max, per cent	....	...	...	...	....	0.25
90 per cent distillation point, max, F	....	...	662	...	....	540–620
10 per cent distillation point, max, F	....	...	...	...	....	425–500
End point, max, F	....	...	...	...	....	675
Flash point, min, F	110	150	140	150	150	150
Pour point, max, F	35	30	14	...	25	−5
Ash, max, per cent	0.02	0.03	0.05	0.05	0.02	0.01
Water and sediment, max, per cent	0.1	0.5	0.1	0.5	....	0.05
Precipitation number	....	...	0.25	...	....	....
Sulphur, max, per cent	1.5	2.0	0.75	2.0	1.75	0.50
Corrosion at 210 F, max	....	...	...	...	....	Tarnish
Acidity	Neutral	...	Neutral	...	....	Neutral
Heat value, Btu per lb, min	....	18750	19000	...	....	....
Specific gravity at 60 F, max	....	...	0.895	...	0.920	....
Specific gravity at 60 F, min	....	...	0.840	...	....	....
A.P.I. gravity, max	....	...	...	...	....	40
A.P.I. gravity, min	....	...	...	...	22.3	32
Water by distillation, max, per cent	....	...	...	...	0.25	....
Sediment by extraction, max, per cent	....	...	...	...	0.1	...

Engine Deposits and Wear. Due to reduced exposure of cylinders to corrosive, intermediate products of combustion, the possibility of more complete burning, the tendency toward elimination of partially burned fuel, and the practical toleration of greater quantities and kinds of deposits and, consequently, wear in large engines, the specifications limiting the maximum amount of impurities become more liberal as the engine size increases and speed decreases.

Availability. It is difficult to make rigid statements with regard to the comparative availability of the four classes of Diesel fuels, as conditions with regard to supply and demand change frequently. In general, the great range in Diesel fuels, from a light kerosene distillate to a heavy cracked residual fuel, greatly increases the availability of Diesel fuels as compared to carburetor-type fuels.

Examples of Diesel-Fuel Specifications

Heater-Type Fuels. Table 6 shows examples of fuels requiring heating and cleaning before use. They are included in the general class of residual or bunker fuels and are characterized by high densities, sometimes heavier than water, high viscosities, 1000 sec SU and above, high carbon residues, high percentages of asphaltic matter, and ash.

Large Slow-Speed-Engine Fuels. Fuel specifications for large slow-speed engines are given in Table 7. These fuels are characterized by large ranges in viscosity, in one case, from 40 to 200 sec SU, large distillation ranges, approximately 375 to 850 F, high percentages of impurities, and ignition qualities down to 30 cetane number. The maximum limits for carbon residue vary considerably from a low value of 0.75 per cent to a high value of 3 per cent. The British fuel shown in Table 7 is of the heater type because of its high viscosity limit and percentages of impurities.

TABLE 9 DIESEL-FUEL SPECIFICATIONS FOR LIGHT HIGH-SPEED ENGINES

Property	A.S.T.M. 1-D proposed 1939	British B.S.I. 1937	Refiner	Manufacturer
Cetane number, min	45	..	...	38–48
Aniline point, min, F	..	140	...	...
Viscosity, SSU at 100 F	..	51 Max	35–45	35–40
Carbon residue, per cent, max	..	0.2	0.1	0.03
Flash point, min, F	100	150	150	150
Sulphur, max, per cent	0.5	1.5	1.0	0.5
65 per cent boiling point, max, F	..	662	...	...
Final boiling point, max, F	590	..	...	700
Ash, max, per cent	0.01	0.01	0.01	0.02
Pour point, max, F	0	20	20	0
Copper-strip corrosion at 122 F	Pass	..	...	...
Alkali and mineral acid	Neutral	..	...	...
Water and sediment, max, per cent	0.05	0.1	0.1	0.2
Heat value, min, Btu per lb gross	..	19000	...	...
A.P.I. gravity at 60 F	..	..	29.3–38	26–29
Color, N.P.A., max	..	..	8	...

TABLE 10 U. S. NAVAL DIESEL-FUEL SPECIFICATIONS

Property	Specification 7-0-2	7-0-2a	7-0-2b	7-0-2c
General fuel type	Hydrocarbon	Distillate	Distillate	Distillate
Cetane number, min	....	....	....	....
Diesel index number, min	....	....	40	45
Viscosity, SSU at 32 F	200 Max	200 Max	....	....
Viscosity, SSU at 100 F	....	....	35 to 55	35 to 45
Pour point, max, F	....	....	0	0
Flash point, closed cup, min, F	150	150	150	150
Carbon residue, max, per cent	....	....	0.5	0.2
Total sulphur, max, per cent	1.50	1.50	1.0	1.0
Ash, max, per cent	0.1	0.1	0.01	0.01
Water and sediment, max, per cent	0.5	0.5	0.05	0.05
Corrosion at 212 F	....	....	....	Negative
90 per cent distillation temperature, F	....	....	....	675
Precipitation number, max	....	....	0.3	....
Color, N.P.A., max	....	....	....	....

Medium-Speed-Engine Fuels. The ranges in viscosity and distillation temperatures for the medium-speed class, Table 8, are definitely less than for the slow-speed classification, ranging from about 35 to 75 sec SU and 350 to 750 F, respectively. The maximum allowable percentage of impurities is reduced and the ignition quality has been raised to a minimum limit of 35 cetane number and above.

High-Speed-Engine Fuels. High-speed-engine fuels range from a light gas oil up into the kerosenes as shown in Table 9. They are typified by narrow ranges in viscosity and distillation temperatures, viz., about 32 to 45 sec SU and approximately 350 to 650 F, respectively. Ash, sulphur, carbon residue, water, and sediment and corrosion are kept as low as is practical from the refining, handling, and cost standpoints. The pour point may be specified at zero, or below, as this type of fuel is often used in services where efficient cold starting and low-temperature pumping are important. The minimum allowable ignition quality has also been increased to an average of about 45 cetane number.

Conclusions

1 The approximate fuel ignition quality, in general, increases with engine speed, viz.:

Heater-type fuel	No limits
Slow-speed fuel	25–35 Cetane number
Medium-speed fuel	35–45 Cetane number
High-speed fuel	45 Cetane number and above

2 The allowable viscosity and distillation ranges become narrower, with lower values, as the size of the engine decreases and its speed increases, viz.:

Type of fuel	Viscosity range, SSU	Distillation range, temp F
Heater-type	200–6000	No limits
Low-speed	40–200	375–850
Medium-speed	35–75	350–750
High-speed	35–45	350–650

3 Carbon residue above approximately 3 per cent may cause marked increases in fuel consumption and exhaust-gas temperatures in relatively large slow-speed-type engines.

4 Salt water, either in the fuel or the intake air, has progressively increased engine wear.

5 Corrosive compounds inherent in the fuel or resulting from refining processes and subsequent incomplete removal have caused corrosion of storage facilities and injection systems resulting in engine failures.

6 Sulphur above approximately 0.7 per cent has caused relatively rapid increases in engine wear.

7 Ash should be held to a practicable minimum in all fuels, but the maximum allowable amount may increase with cylinder size.

8 Sufficient data are not available to show, quantitatively, the effects of volatility, high boiling-point fractions, and carbon residue on the combustion characteristics of a Diesel engine.

9 Sufficient data are not available to make analyses and draw comprehensive conclusions on the effects of sulphur, water, corrosive impurities, ash, incomplete combustion, gum- and lacquer-forming constituents, and cracked products on engine wear.

10 The use of heater-type fuels, i.e., bunker fuel oils, in Diesel engines can be justified, generally, only on the basis of low over-all cost, including possible adverse engine performance and increased maintenance, wear, and casualties which, together with somewhat reduced engine availability, practically eliminate their use for modern, high-specific-power engines in the military services.

11 The large variation in fuel-property specifications existing

within each fuel classification is caused by the large differences in present injection-system and combustion-chamber types and design details, which, in turn, prevent practical, fuel-specification standardization.

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Superheat Control and Steam Purity in High-Pressure Boilers

By MARTIN FRISCH,[1] NEW YORK, N. Y.

The author discusses the problem of designing a high-pressure boiler to deliver steam at high temperatures. The problem becomes more difficult as the capacity increases since proper selection and maintenance of boiler materials, design of the separate boiler and furnace units, and maintenance of the latter becomes increasingly more difficult if superheat control and steam purity are to be maintained. The author analyzes these design features and points out the compromises necessary to meet varying operating conditions. He discusses superheater design, maintenance of heat-absorbing surfaces, boiler circulation, relation of furnace-exit temperatures to slagging and boiler performance, and safe metal temperatures in water-walls and superheaters.

THE problem of designing a high-pressure steam-generating unit to deliver steam at high temperature becomes increasingly difficult as the capacity increases. The most important of the many phases of the problem are:

1 The maintenance of external cleanliness of the heat-absorbing surfaces. (Freedom from slagging.)

2 The maintenance and control of the desired steam temperature over the desired load range.

3 The production of steam of the required purity.

4 The maintenance of safe metal temperatures in the pressure parts of the superheater and boiler.

The first two phases of the problem are difficult to reconcile because with the low furnace-exit temperatures conducive to slag-free operation it becomes difficult, if not impossible, to obtain high steam temperatures over a wide load range with a convection superheater of practical size.

Increases in steam-generator capacities are generally provided for by proportional increases in furnace volume. But slag-free operation with a unit of one size, and coal, does not mean that slag-free operation will follow at the same liberation rate with the same coal if the size and capacity are materially increased, because, at the same rate of heat release in the furnace per cubic foot of furnace volume, the larger the furnace, the higher the furnace-exit temperature. Fig. 1 shows that as the furnace volume increases, there is a rapid decrease in the available furnace perimeter per cubic foot of volume which may be covered with radiant-heat-absorbing surfaces. This ratio may be considered as an index of the cooling capacity of the furnace. For example, if the furnace volume is doubled, the cooling-capacity index decreases 25 to 30 per cent. The furnace-exit temperature, as shown by the upper left-hand curves for a furnace of constant height increases rapidly with furnace volume. Furthermore, if 100 per cent of the available surface perimeter is not covered with heat-absorbing surfaces, or if the surface is installed but made ineffective by slag, the furnace-exit temperature rises rapidly. Thus if 25 per cent of the surface is not effective or not cooled, the furnace-exit temperature will be about 200 F higher than would be the case if 100 per cent of the perimeter is cooled.

If a limit is placed on the maximum permissible furnace temperature, the size of furnace which may be designed to comply is definitely limited. The lower the height of the furnace and the smaller the portion of the furnace perimeter that is cooled, the smaller the size of unit that may be honestly considered to comply, unless (*a*) the heat release is reduced as the capacity is increased in about the same ratio as the furnace-cooling index decreases, or (*b*) the amount of radiant-heat-absorbing surface installed per cubic foot of furnace volume is increased in some suitable manner.

Alternative (*a*) means larger and higher furnaces with lower and lower rates of heat release as the capacity is increased and is the simplest solution if the space is available. If space conditions limit the amount of furnace volume which may be provided, it is possible to subdivide the available furnace volume into two or more furnaces using radiant-heat-absorbing partition walls. Each subdivision adds two such walls with a possible increase of 25 to 30 per cent in the cooled wall surface per cubic foot of furnace volume. This corresponds to a reduction of about 100 F or more in the furnace-exit temperature.

The curves at the right of Fig. 1 indicate the marked increase

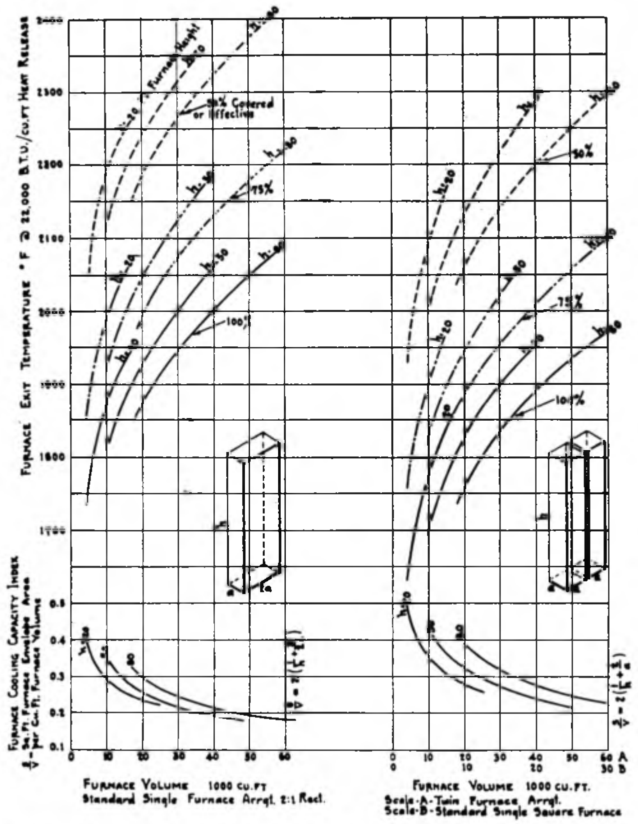

FIG. 1 FURNACE SIZE VERSUS COOLING CAPACITY AND EXIT TEMPERATURE

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Contributed by the Power Division and presented at a meeting, April 23, 1940, of the Pittsburgh, Pa., Section of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

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

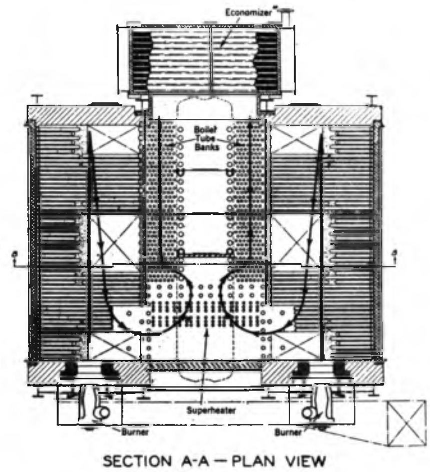
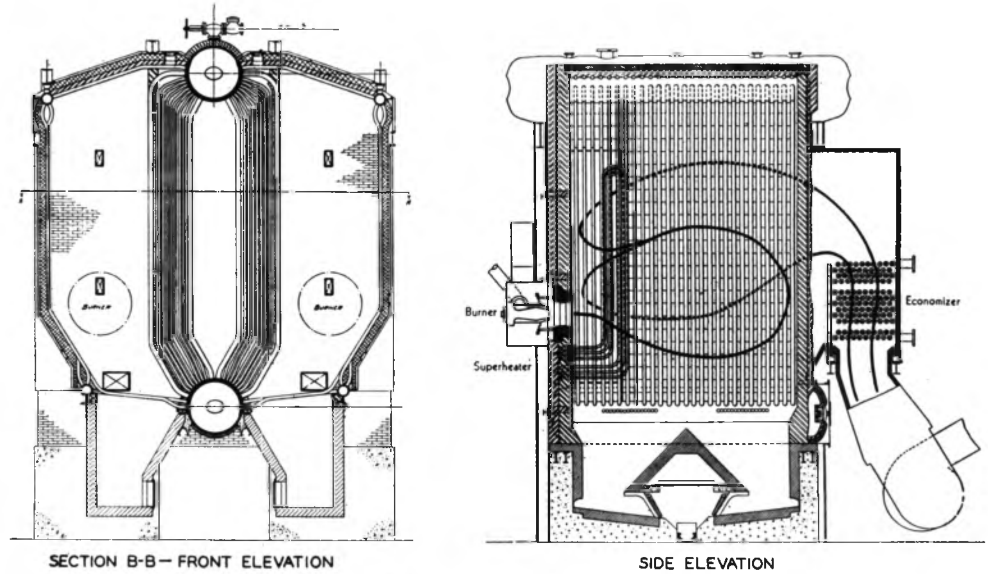


Fig. 3 A Twin-Furnace Unit With Convection Superheater, Having a Capacity of 100,000 Lb of Steam per Hr, Installed at Oil City, Pa.



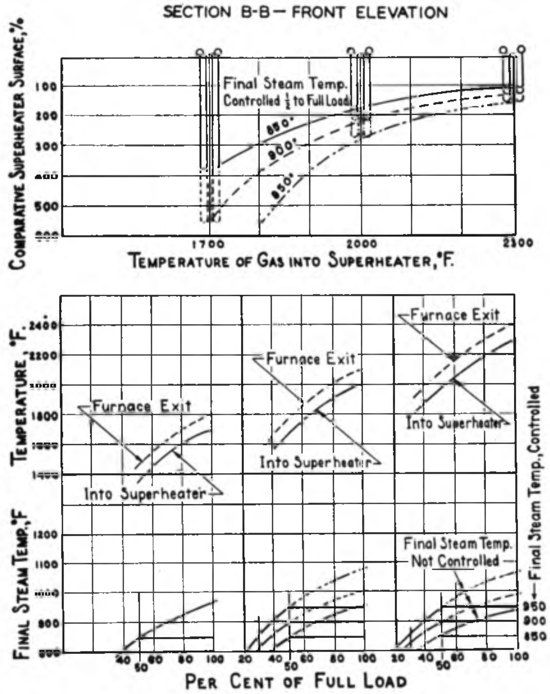


Fig. 2 Furnace-Exit Temperature Versus Superheater Size and Control Range

in available furnace-wall area per cubic foot of furnace volume obtainable by subdividing the furnace into two furnaces. There is a correspondingly lower furnace-exit temperature for a given furnace volume for the same heat release per cubic foot. The same low furnace-exit temperature could be attained at the same capacity in a conventional single-furnace unit by reducing the heat release 25 to 30 per cent or for the examples used from 22,000 to between 16,500 and 15,400 Btu per cu ft per hr.

While low furnace-exit temperatures are desirable to prevent slagging of the boiler and superheater, the permissible temperature reduction is limited by the superheater characteristics desired and by the type of superheater used. If a convection type superheater must produce a very high steam temperature at a very low load, its proportions may be impractical, or uneconomical, unless the furnace-exit temperature is so high that slagging of the superheater becomes certain.

For example, the lower curves of Fig. 2 show typical superheater characteristics obtainable with full-load furnace-exit temperatures of 1800, 2100, and 2400 F and corresponding gas inlet temperatures to superheater of 1700, 2000, and 2300 F.

With an 1800-F furnace-exit and a corresponding 1700-F superheater inlet temperature, it is possible to design a superheater for a constant final steam temperature of 850 F at 1350 psi from half load to full load. With a superheater of prohibitive and impractical size it might be possible to obtain the same control range with a final steam temperature of 900 F. However, it would be impossible to obtain 950 F, except at loads exceeding 70 per cent of full load. If the gas temperature to the superheater is increased to 2000 F, it is possible to obtain 850 F final steam temperature down to about 30 per cent, and 900 F final steam temperature down to 40 per cent of full load. With a gas temperature of 2300 F, it would be possible to obtain 850 F final steam temperature down to 27 per cent, and 900 F final steam temperature down to 40 per cent of full load. The relative cost of superheaters for 850, 900, and 950 F final steam temperature over a load range from one half to full load may be approximated from the upper curves of Fig. 2 which show the comparative surfaces for these various gas temperatures. Assuming as a basis gas at 2300 F and 850 F steam temperature over 50 per cent load range, it may be seen that 60 per cent more surface is required when the superheater inlet gas temperature is 2000 F, and 260 per cent more surface is required when the gas temperature is 1700 F.

To obtain 900 F final steam temperature over a 50 per cent load range, the amount of surface required with a superheater gas-inlet temperature of 1700 F is four times that necessary with 2300 F and 2.6 times the surface adequate at 2000 F.

Obviously, the designer limited to the use of a convection superheater is forced to compromise between low furnace-exit temperature—slag-free operation—and sufficiently high gas temperature for a practical superheater to give the desired steam temperature over the required load range.

Many purchasers who have been, or are, troubled by slagging difficulties arbitrarily specify that new units for them should be designed so that the furnace-exit temperature will be below the lowest expected ash-fusion temperature. In some cases it is even more drastically demanded that the furnace temperature should be less than the initial deformation or agglomeration temperature. For eastern coals this temperature may be as low as 1800 to 1900 F.

Such severe specifications cannot be honestly complied with by conventional units with convection superheaters, especially in large units for high pressures and temperatures if fired with low-grade coals of high ash content and low ash-fusion temperature. If sufficient furnace and radiant-heat-absorbing surfaces can be provided to cool the gases to the desired low value before leaving the furnace, it may be found impossible to obtain the desired final steam temperature over the required load range with any practical convection-superheater arrangement. This difficulty may only be side-stepped by accepting a furnace-exit temperature which may be several hundred degrees higher than originally prescribed, or by using radiant-superheating surface in place of waterwall surface for furnace cooling.

Slagging of superheaters and boiler tubes ahead of them is responsible for a great deal of capacity reduction, outage, and cleaning expense. Periodic or continuous hand lancing is often required to keep such units in service because soot blowers and deslaggers fail to check fouling. Furthermore hand lancing is laborious, disagreeable work, and there may come a time when men willing to do such work will be scarce. It is possible to design steam-generating units in which slag formation may be kept under control and the desired steam-temperature characteristics obtained.

The use of multiple furnaces offers a happy solution of the problem of getting high steam temperatures over a wide load range with low furnace-exit temperatures and comparative freedom from slagging. One of the chief limitations to the use of all superheaters, namely, the difficulty of keeping metal temperatures within safe limits during the starting period, can be eliminated with a multiple furnace arrangement by confining the radiant superheater to one furnace which is fired up only after the boiler starts to generate steam. In such a case, the radiant superheater

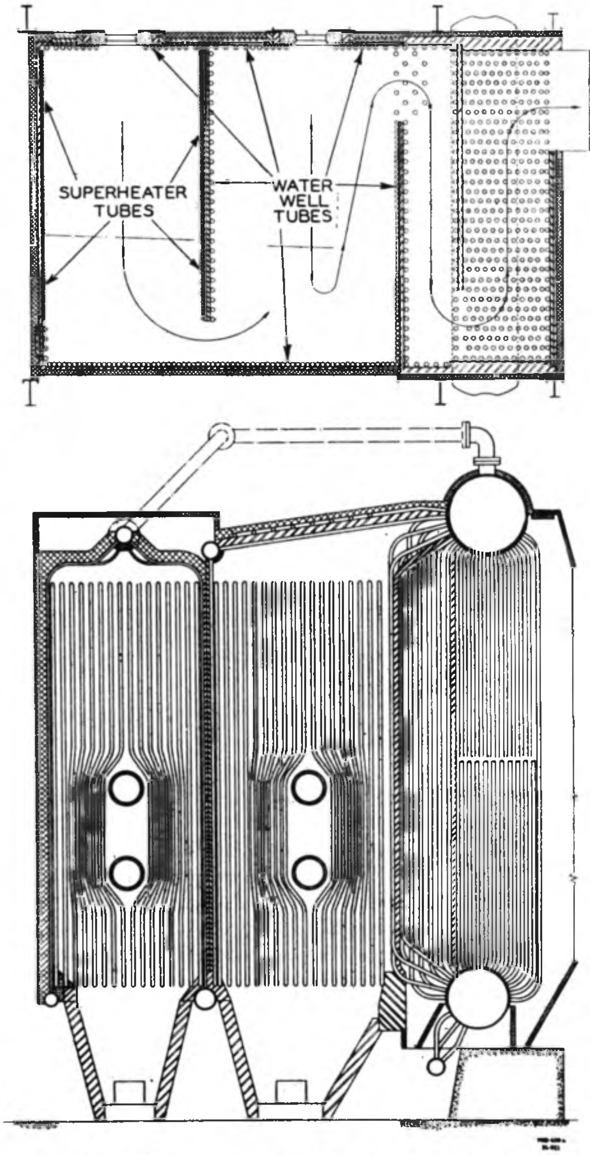


Fig. 4 A Twin-Furnace Unit With Radiant Superheater in One Furnace, Having a Capacity of 120,000 Lb of Steam per Hr, Installed at Moose Jaw, Sask., Canada

is never exposed to fire unless steam is flowing through it.

A number of successful twin-furnace boilers have recently been put into successful operation with and without radiant superheaters. Fig. 3 shows a twin-furnace unit, with only a convection superheater, which has a capacity of 100,000 lb per hr at 725 psi and 750 F final steam temperature. It was installed at Oil City, Pa., in 1935 and was followed by a duplicate order in 1939. Fig. 4 shows a twin-furnace unit at Moose Jaw, Saskatchewan; one furnace is cooled by a radiant superheater while the other is cooled with waterwall tubes only. This unit has a capacity of 120,000 lb of steam per hr at 650 psi and 850 F final steam temperature. It includes no convection superheater. Fig. 5 shows a twin-furnace unit at the Windsor Plant with a radiant superheater in one furnace, in series with a booster convection superheater receiving gases from both furnaces; the capacity is 750,000 lb of steam per hr at 1350 psi and 925 F final steam temperature. The units shown in Figs. 3, 4, and 5 all burn pulverized coal having an ash-fusion temperature often as low as 1900 F.

While the use of multiple furnaces with or without radiant

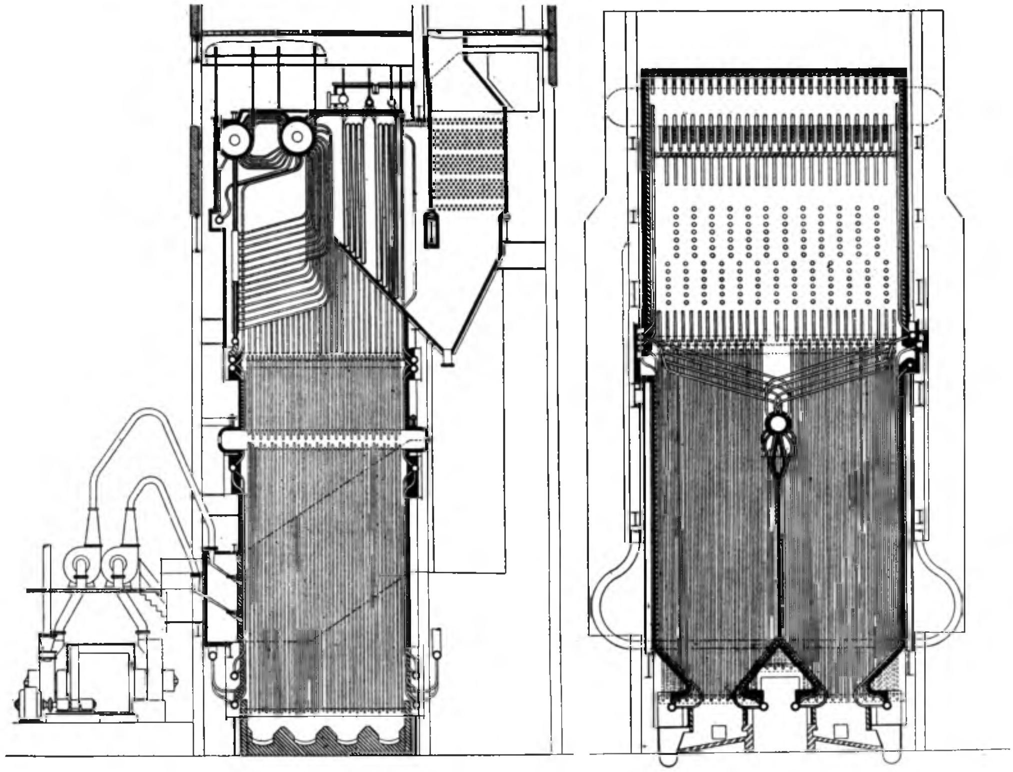


Fig. 5 A Twin-Furnace Unit With Radiant Superheater in One Furnace in Series With Convection Superheater Receiving Gas From Both Furnaces. This Unit Has a Capacity of 750,000 Lb of Steam per Hr and Is Located at Power, W. Va.

superheaters permits the reconciliation of the conflicting objectives of high steam temperatures and low furnace-exit temperatures, there are sometimes obstacles in the way of applying such designs. Then conventional, single-furnace units, as shown in Fig. 6, must be designed to satisfy the operating conditions as well as possible. This unit, located at New Castle, Pa., is one of seven of the type in operation in various parts of the country with a variety of coals. It generates 400,000 lb of steam per hr at 900 psi and 900 F final steam temperature.

Table 1 shows comparatively the furnace characteristics of the units illustrated in Figs. 3, 4, 5, and 6.

Whether or not it is possible to reduce the furnace-exit temperature as much as desired, the slagging tendency may be minimized by fine pulverization. Fine particles burn faster than coarse particles and may be expected to contain less unburned fuel residues when they pass out of the flame into the clear gas space between the flame and the furnace exit. Such particles can then give up heat by radiation to cold surrounding wall surfaces. If the particle path with respect to the furnace cooling surfaces and the flame is correct, the rate of heat dissipation by radiation from the particle will exceed the rate of heat reception by the particle by radiation from the flame and by convection from the

TABLE 1 FURNACE CHARACTERISTICS OF BOILERS SHOWN IN FIGS. 3, 4, 5, AND 6

Fig. no.	3	4	5	6
Capacity, lb per hr	100000	120000	750000	400000
Pressure, lb per sq in.	725	650	1555	900
Final steam temperature, F	750	850	925	900
Heat release, Btu per cu ft per hr	21200	20200	23700	24500
Furnace volume V, cu ft	6280	8700	40000	21950
Furnace perimeter S, sq ft	2875	3390	9174	5378
Furnace cooling-capacity index, S/V	0.458	0.389	0.229	0.308
Max cooling surface possible per cu ft, Sm/V	0.720	0.610	0.360	0.490
Actual cooling surface installed per cu ft, Sr/V	0.390	0.560	0.340	0.390
Coverage, per cent	54	92	94	78
Height of furnace, ft	34	34	72	62
Furnace-exit temperature:				
At 100 per cent coverage	1700	1720	1900	1950
At actual coverage	1830	1740	1930	2000
Ash-fusion temperature	1900–2300	2000–2100	1900–3000	2200–2600

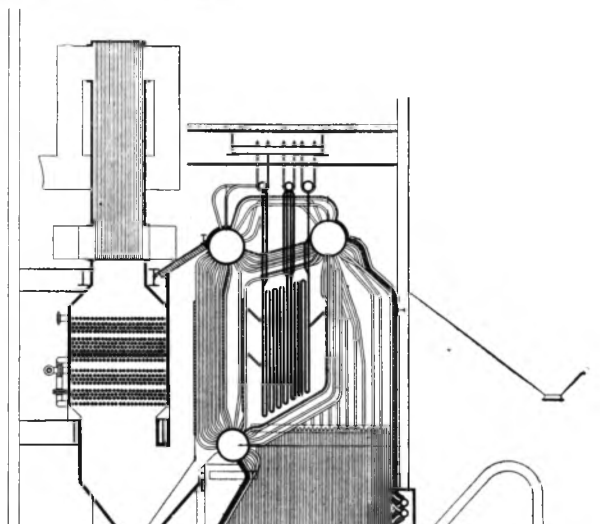


Fig. 6 A Standard Single-Furnace Unit Designed for Burning Pennsylvania Coal. It Is Installed at New Castle, Pa., and Has a Capacity of 400,000 Lb of Steam per Hr

surrounding gases. The smaller the particle, the more certain it is to be frozen throughout by the time it passes into the boiler and superheater.

The additional cost of grinding fine will be partially offset by fuel savings due to reduced carbon loss and more than offset by the reduced cost of boiler cleaning and savings due to fewer outages and resultant loss of production. For example, experience indicates that in the average installation burning Pittsburgh coal one may expect the relationship between fineness, fuel loss due to incomplete combustion, and power consumption and maintenance for a ball-mill job, to be approximately as given in Table 2.

TABLE 2 RELATION BETWEEN FINENESS OF PULVERIZED COAL, FURNACE PERFORMANCE AND MAINTENANCE

Coal fineness, per cent through 200 mesh...	65	70	80	90
Fuel loss due to incomplete combustion, per cent of fuel fired........................	2.0	1.8	1.4	1.0
Fuel loss, lb per ton burned...............	40	36	28	20
Power consumption, kwhr per ton..........	13	15	21	30
Coal equivalent of power at 1.5 lb per kwhr	19.5	22.5	31.5	45
Maintenance cost, cents per ton...........	0.9	1.0	1.4	2.0
Coal equivalent of maintenance cost for coal at $5 per ton.........................	3.6	4.0	5.6	8.0
Net coal equivalent of carbon loss + power consumption + maintenance...........	63.1	62.5	65.1	73.0
Per cent of coal fired......................	3.16	3.12	3.26	3.65

Under these conditions the minimum coal consumption occurs at a fineness between 70 and 75 per cent through 200 mesh. If the power consumption or its coal equivalent is lower, or the carbon loss higher, the economical fineness is higher. Actually one may safely conclude that considerable increases in fineness may be made without materially increasing the over-all fuel cost. If the savings due to reduced slagging, reduced boiler-cleaning expense, outage, and loss of production are considered, increased fineness will actually be found to result in lower over-all costs of steam production. The value of fineness should really be evaluated in terms of its effect on slagging, unit reliability, and availability. Fig. 9 shows the effect of fineness on the over-all cost of making steam, in terms of the fuel used for three sizes of plant, if cleaning crews, as indicated, receiving $40 per week per man are used.

While steam-generator reliability and availability depend to a large extent on adequate provisions for slag control, the acceptance of a unit depends also on its ability to deliver steam of extremely high purity.

Steam which is to all intents chemically pure is now required by nearly every steam-generator specification in order to minimize turbine-capacity loss due to blade deposits or hazards due to turbine control-valve fouling. The problem of producing steam of such purity is tied up with boiler circulation, water-level control, and feedwater conditioning. The problem of feedwater conditioning will not be discussed in this paper. Nor can the design problems involved in providing a safe circulating system be completely discussed in the space available. However, it is well to review general principles governing natural circulation. These are as follows:

(*a*) Circulation is easier to maintain in vertical or nearly vertical tubes than in horizontal or slightly inclined tubes.

(*b*) The water flow to be provided per tube in order to maintain it in a furnace at the highest rate of heat input likely to occur is based entirely on experience which indicates that an internally clean tube will not fail if the entering velocity of steam-free water is over 1 fps, and the ratio of water by volume leaving a tube is not less than 5 per cent.

(*c*) The total circulation in a circulating system suitable for high-pressure operation increases as the operating pressure decreases. The system, therefore, may be safely operated at lower pressures up to limiting capacities which vary with the pressure. The superheater pressure drop, rather than circulation, limits the capacity attainable at low pressure for, except at very low pressures, the circulation is good enough to permit operation of the boiler and waterwall at capacities up to or in excess of the maximum capacity at the design pressure.

(*d*) The total circulation decreases slowly with decrease in output, except at extremely low loads, when the decrease becomes more rapid.

(*e*) The total circulation in a given system is a maximum if the water supplied to the circulating system is free of steam. Therefore, it is important to so design the system that the waterwall supply tubes or downtakes entrain as little steam as possible.

Fig. 7 summarizes a circulation analysis of a completely water-cooled steam-generating unit designed to deliver 300,000 lb of steam per hr at 1375 lb per sq in. at the superheater outlet with 1450 lb per sq in. in the drum. Three groups of curves show the characteristics of the waterwall circulation system at the full drum operating pressure of 1450, 850, and 200 lb per sq in. abs, respectively. The dashed curves show the effect of steam entrainment of as much as 25 per cent by volume by the downcomers. This amount of steam entrainment reduces the total circulation materially but not sufficiently, except at a drum pressure of 200 lb per sq in., to limit the safe capacity of the system. A capacity limitation at this low pressure would be advisable because of the reduction in amount of water by volume at the discharge ends of the front wall tubes.

A high-pressure unit would not ordinarily be operated at such low pressure and high capacity because superheater pressure drop would increase rapidly with decrease in superheater outlet pressure. Nor would a good design lack provisions to prevent steam entrainment by the water-supply tubes. Therefore, the real significance of the curves is that the tubes of a properly designed circulating system would be safe under most conceivable operating conditions.

The rate of circulation in the furnace waterwall system has a controlling effect on the behavior of the water levels in the boiler and on the problem of steam purification.

Since the bulk of the steam produced by the steam-generator unit is made by the waterwall system and the first four rows of the boiler, the bulk of the water circulation will be in these elements. Only a small fraction of the total steam produced and a correspondingly small portion of the total circulation will occur in the other boiler tubes. In view of this, if only the steam produced by these relatively inactive tubes is delivered to the rear drum below the water line, the turbulence in this drum from this source will be slight and the density of the water-steam emulsion below the water line relatively high. Therefore, it is usual practice to locate the steam purifier, through which the steam passes on its way to the superheater, in this the most quiet drum Q and to limit turbulence in the drum by connecting all waterwall risers or discharge tubes to drum T in which the turbulence is naturally greater. This means that the bulk of the circulating water flows into the more turbulent drum T first where a preliminary separation of steam and water is induced. The steam flows to the quiet drum Q through the steam circulators SC. The problem is to return the water to the circulating system. This is accomplished by taking it through the water circulators WC into the quiet drum Q whence it flows to the lower drum D and thence through the downtakes DT to the waterwall system.

The head for causing the water to flow through the water circulators is made up of two elements:

a = The head corresponding to the pressure difference between drums T and Q, and

b = the elevation of the level in T above the level in Q.

Element a increases approximately as some power (about 2) of the load.

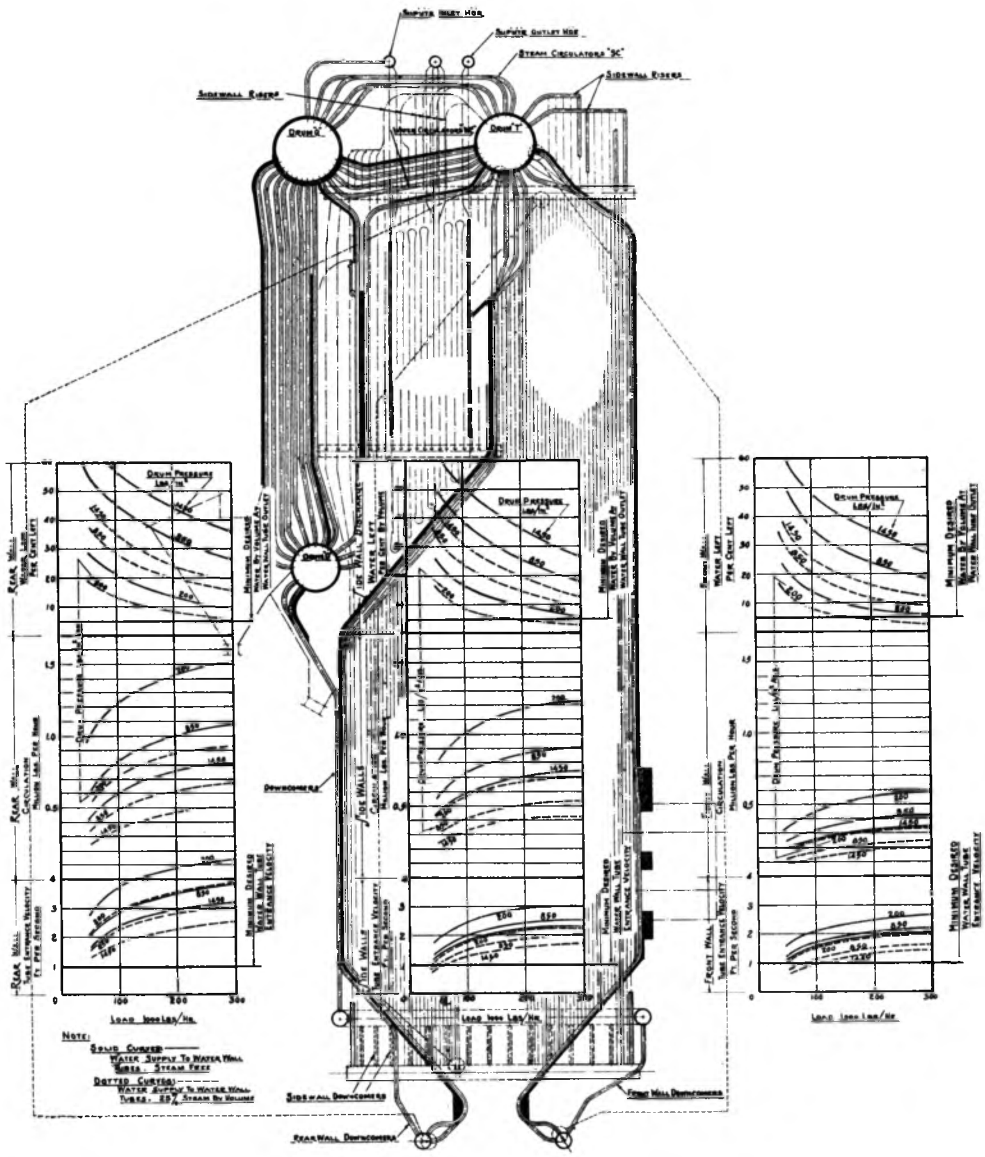


FIG. 7 CIRCULATION ANALYSIS OF A THREE-DRUM HIGH-PRESSURE STEAM GENERATOR AT VARIABLE-PRESSURE OPERATION

Element b depends on the magnitude of a, the density of the water in T and Q, the total amount of water delivered to drum T by the circulatory system and which must be returned to the quiet drum and thence to the circulating system. If the steam circulators were infinite in number and a were zero, b would be independent of a. If the value of b under these conditions is indicated by b_0, then

$$b = b_0 - a$$

The value of b_0 as shown by Fig. 8 changes slowly with load since the total circulation as may be seen by Figs. 7 and 8 does not change very rapidly with load above a certain low load.

Fig. 8 shows approximate relationship between b, b_0, a, and the load.

The elevation of the water in the turbulent drum varies with load as shown by the curve b. Obviously, it is desirable that at no load should the elevation b rise to the level of the lowest steam circulator. This is less important from the point of view of carry-over (if an adequate purifier is installed in the other drum) than from the point of view of having a steady water level. If the level b is allowed to rise above any of the steam circulators, these become water circulators and the number of steam circulators is correspondingly reduced. If one row of steam circulators out of three become water circulators and cease to become steam circulators, the steam velocity in the circulators increases by 50 per cent and a increases approximately 125 per cent. The water-circulator capacity is correspondingly increased and the effect is to instantly depress the level in the turbulent drum. As soon as the level is depressed below the steam circulators the original pressure and head conditions are re-established and the level starts to rise again. This process repeats itself indefinitely and causes the often observed swinging water level. This may be corrected in five ways:

1 Raise the turbulent drum.

2 Lower the controlled level in the quiet drum.

3 Increase the number of water circulators or take some of the water supply for the waterwall system directly from the turbulent drum so that the water circulators will not have to handle it.

4 Decrease the number of steam circulators.

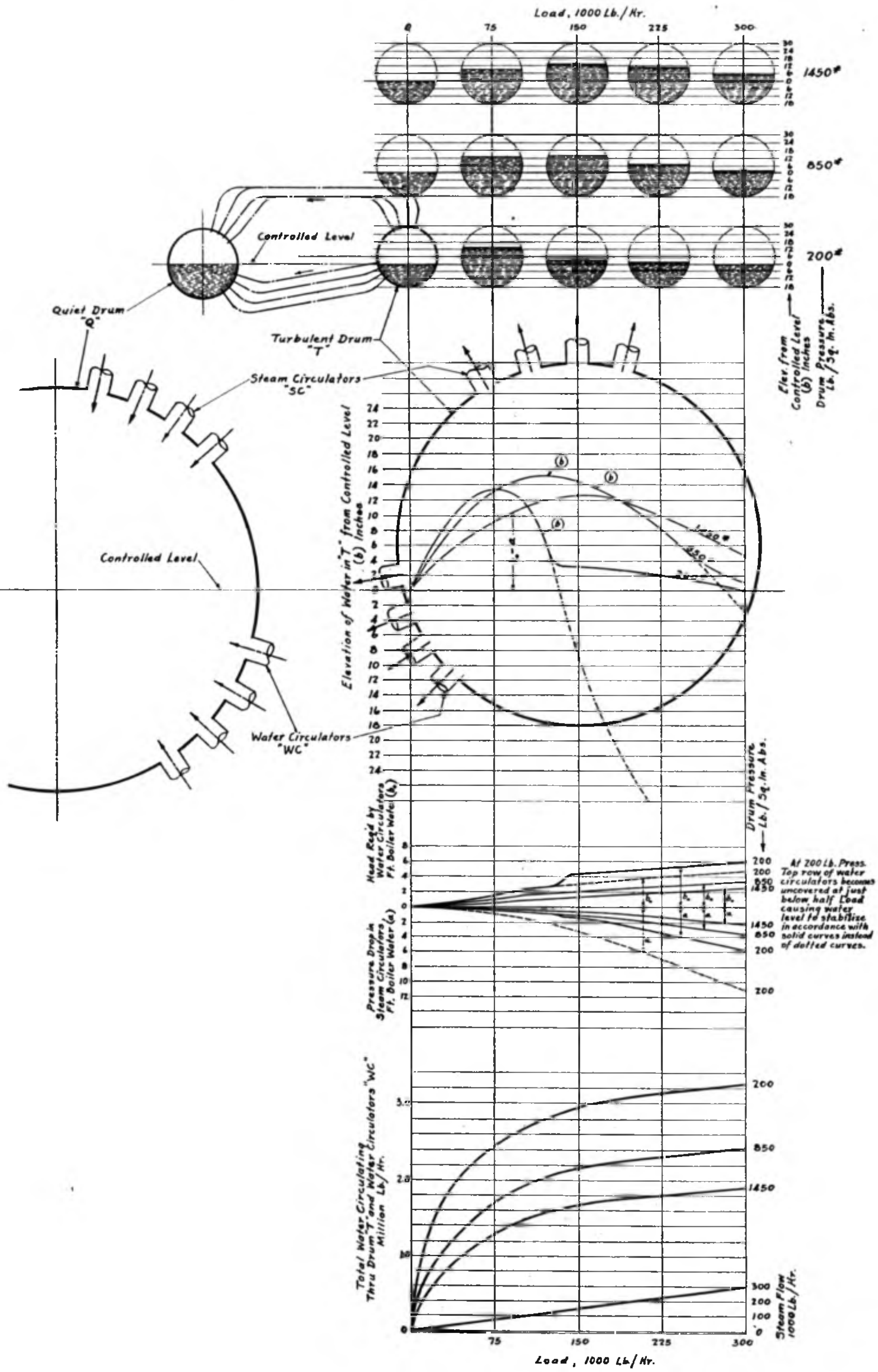


FIG. 8 STUDY OF WATER-LEVEL BEHAVIOR IN A THREE-DRUM HIGH-PRESSURE BOILER AT VARIABLE-PRESSURE OPERATION

5 Arrange the waterwall discharge tubes to have as many as possible discharge above the water line in the turbulent drum, thus making the density of the steam-water mixture below the water line as great as possible.

Of these five correctives, the fourth one does not have as much effect as one might expect. Tests of a number of boilers have borne this out. The reason for this may be seen by referring to Fig. 8 which makes it clear that the effect of increasing the pressure drop between drums by decreasing the number of steam circulators becomes marked only at very high loads. On the other hand the maximum elevation of the level in the turbulent drum above that in the quiet drum occurs near half load when the pressure drop *a* through the steam circulators is small compared with the head required to cause the circulating water to be transferred from the turbulent drum *T* to the quiet drum *Q* through the water circulators *WC*.

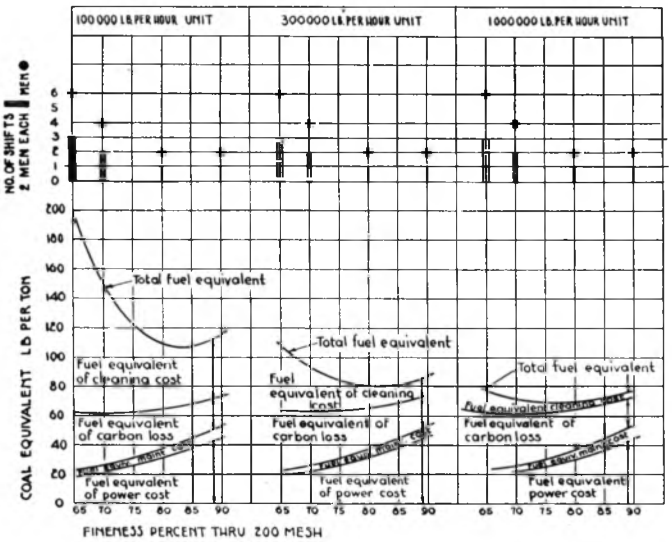


Fig. 9 Effect of Fineness on the Over-All Cost of Making Steam

Most three- and four-drum bent-tube boilers built in the past and still being built are not equipped with an adequate number of water circulators between drums. In this may be found an explanation, at least in part, for the absence of water in the back drum of a four-drum boiler at high loads and for fluctuating water levels and the swamping of the front drum of most three-drum boilers at intermediate loads.

The desirability of elevating the turbulent drum with respect to the quiet drum a reasonable amount and of connecting this drum to the quiet drum with as many water circulators as possible is self-evident.

Steam of uniformly high purity under all conditions is more certainly obtainable if the water levels behave in a predictable manner. Too much emphasis cannot be placed on the necessity of having the waterwall and boiler circulation properly coordinated to this end.

Having taken steps to establish proper water-level behavior, steam of high purity may be obtained by washing or by the use of moisture-eliminating devices which, if the boiler-water concentration is very high, must be nearly 100 per cent effective. With steam washers, since the entrained moisture contains smaller quantities of impurities than the boiler water, the drying of the steam need not be as effective.

Steam washers work on the principle of exchanging for the very impure boiler water, feedwater of much higher purity and then subjecting the mixture of steam and entrained feedwater to a drying operation. One effective exchange method is to bubble the steam through the feedwater at extremely low velocities. The exchange efficiency of this type of washer is between 80 and 90 per cent, and steam with as little as one part per million may be obtained with a boiler-water concentration of 2500 ppm using a feedwater for washing which contains up to 200 ppm of impurities. Certain types of feedwater treatment make the advisability of steam washing questionable because of the tendency of the washer to sludge up too rapidly. Under such conditions the washer should be easily accessible and cleanable so that it may be washed out periodically. It is not advisable to use washers for extremely low pressure because of the difficulty of providing sufficiently low steam velocities through the washer.

When the steam delivered by the boiler is of the purity required by the turbine, safe metal temperatures in the pressure parts of the superheater will be a natural consequence if the superheater design gives sufficiently high steam flow through the tubes. For a 2-in. tube the flow should be in excess of 2000 lb per tube per hr and preferably of the order of 3500 to 4000 lb per hr. This makes it necessary and wise to allow for a reasonably high pressure drop through the superheater. For a 1200 to 1350-lb per sq in. job the pressure drop should certainly not be less than 50 lb per sq in., and if the unit is to be operated at very low loads for considerable periods the pressure drop might better be 75 to 100 lb per sq in. in order to assure good distribution at the lowest load. For example in a unit operating at one-fifth load the superheater would have a pressure drop of only 4 lb per sq in. if the full load pressure is 100 lb per sq in.; at one-tenth load the drop would be 1 lb per sq in.

The metal temperatures in the pressure parts of the boiler and waterwall system will be safely low if the circulation is satisfactory and obeys the criteria previously outlined.

Conclusions

The chief problem in the design of steam generators of high reliability especially for high-pressure and high-temperature operation in large sizes is the proper reconciliation of the opposed aims, low furnace-exit temperatures with freedom from slagging and high steam temperatures over a wide range with a practical superheater.

High fineness of pulverized coal helps to minimize the slagging problem as does the use of multiple furnaces with radiant or combination superheaters.

Freedom from circulation troubles can be attained by proper design of the circulating system. This involves the use of an adequate number of waterwall supply and discharge tubes so distributed that all parts of the system receive their proper share of the circulating water as free of steam as possible and utilize a proper share of the available pumping head. Internal cleanliness of the tubes must not be overlooked.

Steam of high purity may be obtained more easily if the circulating system is so designed as to assure proper behavior of the water in the boiler drums so that the maximum amount of steam space is always available.

Superheater-metal temperatures will be safe if the steam is pure and the steam flow rate through the superheater elements is sufficiently high. Waterwall-metal temperatures will be safe if internal cleanliness of the tubes is maintained by adequate water treatment and periodic cleaning and if the circulatory system is of proper design.

Properties of Hydrogen Mixtures

By A. W. BRUNOT,[1] LYNN, MASS.

With the increased use of hydrogen as a cooling medium in rotating electrical machines, it has become important to know the properties of hydrogen diluted by small amounts of another gas. Using these properties it is possible to predict the operation of the machine in a hydrogen mixture from the results of tests made in air or pure hydrogen. It is the purpose of this paper to present a summation of data on the properties of hydrogen, air, nitrogen, and carbon dioxide, as well as empirical formulas for the properties of the mixture of hydrogen with any of the other three gases, and to show how accurately the performance of a heat exchanger in hydrogen or hydrogen mixtures may be predicted from tests in air.

Nomenclature

The following nomenclature is used in the paper:

- p = pressure, lb per sq in. abs
- t = temperature, F
- ρ = density, lb per cu ft
- c = specific heat at constant pressure, Btu per lb per deg F
- μ = viscosity, lb per ft per hr
- k = thermal conductivity, Btu per ft per deg F per hr
- $(1-y)$ = fraction by volume of hydrogen in a binary mixture
- y = fraction by volume of second gas in the binary mixture
- h = local gas-side heat-transfer coefficient, Btu per sq ft per deg F per hr
- G = weight velocity, lb per sq ft per hr
- ΔP = pressure drop, in. water

General Conditions

In this study of the properties of hydrogen diluted by small amounts of another gas, since the pressures encountered in hydrogen-cooled machines are generally less than 2 atm, the effect of pressure on specific heat, viscosity, and conductivity has been neglected. The relations given are for atmospheric pressure but variation in pressure causes little change in properties except density for which a pressure term is included.

In hydrogen-cooled machines the per cent by volume of the second gas is small; therefore, the range of mixtures over which information is desired is 0 to 15 per cent by volume of the second gas. Fig. 1 shows the variation of the properties of mixtures of hydrogen and nitrogen at 70 F as a function of the composition. From the curves it is evident that most of the variation takes place as the first 15 per cent of the second gas is added. The rapidity of the variation, as the heavier gas is added to the hydrogen, may be due to the wide difference in molecular weights but, whatever the cause, the result is that the presence of small amounts of a heavy gas in hydrogen causes a marked change in properties and, hence, a change in the operating characteristics of the fans or heat-transfer equipment.

Due to the scarcity of data on the subject of tertiary mixtures, there has been no attempt made to cover such mixtures in this paper. Instead the paper will deal only with binary mixtures of hydrogen with air, nitrogen, or carbon dioxide.

Density. The density of the individual gases may be calculated directly from the perfect-gas laws. The densities thus computed check very closely the values reported in the International Critical Tables and are the basis for the formulas to be presented.

The density of a mixture of two gases may be calculated directly from Dalton's law. The author has found it more convenient to use, instead of the actual density of the mixture, a density factor, which is merely the ratio of the density of the mixture to the density of pure hydrogen under the same conditions of temperature and pressure.

The density of pure hydrogen is given by the relation

$$\rho_H = 0.188 \frac{p}{t + 460}$$

The density factors are as follows:

Hydrogen-air.............$\rho_m/\rho_H = 1 + 13.36y$
Hydrogen-nitrogen.........$\rho_m/\rho_H = 1 + 12.90y$
Hydrogen-carbon dioxide...$\rho_m/\rho_H = 1 + 20.83y$

Specific Heat. The data on the specific heat of gases at constant pressure are quite plentiful and the results of various investigators check quite closely. The most recent and reliable data have been published by G. B. Taylor (1)[2] and W. M. D. Bryant (2). Their formulas hold throughout a range of 0 to 3500 F with a maximum deviation of 1 per cent for hydrogen and 3 per cent for nitrogen and carbon dioxide. The data for hydrogen, nitrogen, and carbon dioxide were taken directly from these sources, while the data for air were calculated from Bryant's data on nitrogen and oxygen, assuming the air to be 20.9 per cent oxygen and 79.1 per cent nitrogen by volume.

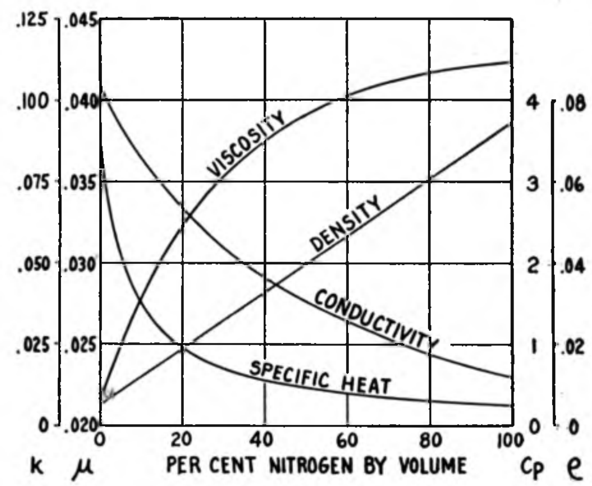


Fig. 1 Properties of Hydrogen-Nitrogen Mixtures at 70 F

The specific heat of a gas mixture appears to obey the well-known law of mixtures, that is

[1] Thomson Laboratory, General Electric Company. Jun. A.S.M.E.

Contributed by the Heat Transfer Group and presented at the Spring Meeting, Worcester, Mass., May 1–3, 1940, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

$$c_m = w_A c_A + w_B c_B$$

where w_A and w_B are the fractions by weight of the two gases present and c_A and c_B their specific heats. The specific heats of the pure gases and the mixtures are given by the following relations:

Hydrogen....... $c_H = 3.430 + 5.75 \times 10^{-5}t + 4.27 \times 10^{-8}t^2$

Air............. $c_{air} = 0.2341 + 3.44 \times 10^{-5}t - 0.463 \times 10^{-8}t^2$

Nitrogen........ $c_N = 0.2407 + 3.26 \times 10^{-5}t - 0.380 \times 10^{-8}t^2$

Carbon dioxide.. $c_{CO_2} = 0.2016 + 9.18 \times 10^{-5}t - 1.736 \times 10^{-8}t^2$

Hydrogen-air.... $c_m = \dfrac{1}{1 + 13.36y}[3.430 - 0.068y + (5.75 + 43.65y)10^{-5}t + (4.27 - 10.92y)10^{-8}t^2]$

Hydrogen-nitrogen.. $c_m = \dfrac{1}{1 + 12.90y}[3.430 - 0.084y + (5.75 + 39.56y)10^{-5}t + (4.27 - 9.55y)10^{-8}t^2]$

Hydrogen-carbon dioxide........ $c_m = \dfrac{1}{1 + 20.83y}[3.430 + 0.971y + (5.75 + 194.65y)10^{-5}t + (4.27 - 42.17y)10^{-8}t^2]$

Viscosity. The available data on the viscosity of pure gases were investigated; the values used as a basis for this paper being selected from the International Critical Tables (3). Formulas to fit these data over the range of 0 to 250 F were then determined.

A search of the literature for a means of calculating the viscosity of a mixture of gases showed that several different formulas had been proposed. Gille (4), working with helium and hydrogen, referred to the work of Puluj and Theisen on the viscosity of mixtures. Trautz (5) proposed a different formula and Schröer (6) and Adzumi (7) give semitheoretical methods of determining the constants in the equation of Theisen. Application of these equations to data on hydrogen-nitrogen mixtures indicated that the equation of the form proposed by Theisen best fitted the data. This equation is of the form

$$\mu_{12} = \frac{\mu_1}{1 + A_1\left(\dfrac{y}{1-y}\right)} + \frac{\mu_2}{1 + A_2\left(\dfrac{1-y}{y}\right)}$$

where y is the fraction of volume of gas 2, and A_1 and A_2 are constants. It was found that the constants in the equation as determined by the method of Schröer (6) and Adzumi (7) fitted the data fairly well and included a term for temperature. However, since the relations proposed by Schröer (6) and Adzumi (7) necessitate an empirical determination of one term and since the variations of the constants with temperature are small over the range of 0 to 250 F, the author chose to determine empirically the constants at an intermediate temperature and apply them over this temperature range. Table 1 shows the variation in the values obtained for the viscosity of hydrogen-nitrogen mixtures, using the methods of Puluj, Trautz, Schröer, and the equation of Theisen with empirical constants as determined by the author.

TABLE 1 VISCOSITY OF HYDROGEN-NITROGEN MIXTURES

Temp, F	Nitrogen, per cent	—Ratio of calculated to reported value—			
		Puluj	Trautz	Schröer	Author
66	19.23	1.0482	1.1454	1.0301	0.9999
66	33.28	1.0329	1.0995	1.0369	1.0034
66	49.47	1.0139	1.0522	1.0320	1.0000
66	79.79	1.0048	1.0140	1.0193	1.0033
212	20.49	1.0586	1.1599	1.0528	1.0136
212	32.81	1.0355	1.1078	1.0319	1.0073
212	49.47	1.0171	1.0571	1.0334	1.0042
212	81.95	1.0012	1.0105	1.0132	1.0003

The table shows the ratio of the calculated value to the value reported in Tables Annuelles de Constantes et Données Numeriques (8).

Rammler and Breitling (9), in an article dealing with the viscosity of flue gases, mention two approximate equations for the viscosity of a mixture but application of these equations to mixtures of hydrogen and nitrogen shows a variation from the data reported in the Tables Annuelles de Constantes et Données Numeriques (8) which is comparable to that obtained using the method of Trautz.

It will be noted from Table 1 that the variation between the calculated and reported values increases as the per cent of nitrogen is decreased in all cases except the author's formulas. Since the region in which we are interested is the region of small amounts of the second gas, the author has in each case determined the constants to be used in the equation by direct substitution of the selected data and these are the values reported.

The data used in determining the empirical constants for the hydrogen-nitrogen mixture were taken from the Tables Annuelles de Constantes et Données Numeriques (8), while those for the hydrogen-carbon-dioxide and hydrogen-air mixtures were taken from the International Critical Tables.[3]

The viscosity of the pure gases and their mixtures are given by the following relations:

Hydrogen.......... $\mu_H = 0.02035\left(\dfrac{t + 459.6}{491.6}\right)^{0.695}$

Air................ $\mu_{air} = 0.04136\left(\dfrac{707.6}{t + 675.6}\right)\left(\dfrac{t + 459.6}{491.6}\right)^{3/2}$

Nitrogen........... $\mu_N = 0.04259\left(\dfrac{732.4}{t + 658.7}\right)\left(\dfrac{t + 459.6}{533.3}\right)^{3/2}$

Carbon dioxide..... $\mu_{CO_2} = 0.03361\left(\dfrac{923.6}{t + 891.6}\right)\left(\dfrac{t + 459.6}{491.6}\right)^{3/2}$

Hydrogen-air. $\mu_m = \dfrac{\mu_H}{1 + 1.803\left(\dfrac{y}{1-y}\right)} + \dfrac{\mu_{air}}{1 + 0.345\left(\dfrac{1-y}{y}\right)}$

Hydrogen-nitrogen.. $\mu_m = \dfrac{\mu_H}{1 + 1.529\left(\dfrac{y}{1-y}\right)} + \dfrac{\mu_N}{1 + 0.385\left(\dfrac{1-y}{y}\right)}$

Hydrogen-carbon dioxide... $\mu_m = \dfrac{\mu_H}{1 + 1.704\left(\dfrac{y}{1-y}\right)} + \dfrac{\mu_{CO_2}}{1 + 0.265\left(\dfrac{1-y}{y}\right)}$

Fig. 2 shows a comparison of the viscosity as calculated from the foregoing equations and the values reported for hydrogen-nitrogen mixtures in reference (8) and for hydrogen-air and hydrogen-carbon-dioxide mixtures in the International Critical Tables.[3]

Thermal Conductivity. The experimental data on the thermal conductivity of gases were scanty and quite scattered as to value, probably due to the difficulty of measurement.

The available data for the last 20 years or so were plotted and showed a marked increase in the value for any particular gas over this time. The amount of increase was 4 to 5 per cent and was definitely upward for all gases, hence, the most recent data were selected for use in this paper. While it is common to see the formula for thermal conductivity of pure gases written in a form

[3] Bibliography (3), pp. 5 and 6.

using Sutherland's constant, the author could not find enough variation from a straight line to warrant the use of any formula but the straight line $K = k_0(1 + \alpha t)$, hence this is the formula used in calculating the conductivity of the pure gas. The equations reported for the pure gases do not strictly follow any particular set of data but rather represent an engineering estimate of the probable value over the range of 0 to 250 F.

There were few data on the conductivity of gas mixtures and no formula was found in the literature to fit the data. However, the same type of equation as used for the viscosity of a mixture was found to fit the data very well if the constants were empirically determined from two test points.

There were available no data on the thermal conductivity of a hydrogen-air mixture. There were, however, data on hydrogen-nitrogen[4] at 32 F and hydrogen-oxygen[5] mixtures at 72 F. Since

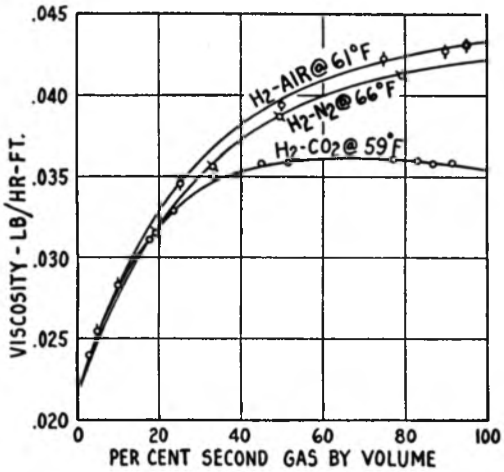


Fig. 2 Viscosity of Hydrogen Mixtures; Comparison of Calculated and Reported Values

air is nearly a 4:1 mixture of nitrogen and oxygen, an attempt was made to predict the conductivity of hydrogen-air mixtures from these data. The constants for mixtures of hydrogen-nitrogen and of hydrogen-oxygen were determined empirically from the data mentioned. Calculated curves for these two mixtures were then drawn at 70 F and the curve for the hydrogen-air mixtures drawn between them. Because of the 4:1 ratio of nitrogen to oxygen in air, the estimated curve for hydrogen-air mixtures was drawn so that the distance from the hydrogen-oxygen curve was $^4/_5$ the distance between the hydrogen-nitrogen and hydrogen-oxygen curves.

The data on the hydrogen-carbon-dioxide mixtures were taken from Physicalish-Chemische Tabellen (10).

The formulas for the thermal conductivity of the pure gases and their mixtures are given as follows:

Hydrogen.......... $k_H = 0.0942(1 + 0.00148t)$

Air................ $k_{air} = 0.0133(1 + 0.00170t)$

Nitrogen........... $k_N = 0.0130(1 + 0.00170t)$

Carbon dioxide...... $k_{CO_2} = 0.00783(1 + 0.00266t)$

Hydrogen-air....... $$k_m = \frac{k_H}{1 + 2.70\left(\frac{y}{1-y}\right)} + \frac{k_{air}}{1 + 0.243\left(\frac{1-y}{y}\right)}$$

Hydrogen-nitrogen.. $$k_m = \frac{k_H}{1 + 2.90\left(\frac{y}{1-y}\right)} + \frac{k_N}{1 + 0.280\left(\frac{1-y}{y}\right)}$$

Hydrogen-carbon dioxide.... $$k_m = \frac{k_H}{1 + 2.85\left(\frac{y}{1-y}\right)} + \frac{k_{CO_2}}{1 + 0.200\left(\frac{1-y}{y}\right)}$$

Fig. 3 shows a comparison between the values as calculated using these equations and the values as reported[4] for hydrogen-nitrogen mixtures and for hydrogen-carbon-dioxide mixtures in reference (10).

Prandtl Number. One of the common dimensionless groups

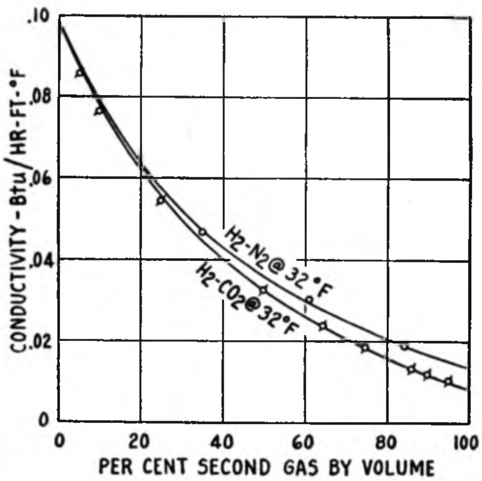


Fig. 3 Conductivity of Hydrogen Mixtures; Comparison of Calculated and Reported Values

used in heat-transfer work is the Prandtl number $c\mu/k$. According to the kinetic theory of gases, this group is a constant for all gases having the same number of atoms per molecule. This is true for ideal and pure gases but there is a variation in the value for impure gases. It was believed that, when two diatomic gases were mixed, this constant remained the same for all compositions, since the value was the same for either of the pure gases alone. Fig. 4 shows a plot of this constant as a function of composition for the three gas mixtures treated in this paper. It will be noted that the "constant" is far from a constant and varies nearly 50 per cent from its value for a pure gas.

The value of the Prandtl number for the pure gases, as determined from the data given in this paper, will vary slightly with temperature instead of remaining constant. This variation is due to the manner in which the equations for the various properties are expressed. The variation over the range of 0 to 250 F is less than 0.5 per cent and for all practical purposes this is negligible.

Application of Data. To illustrate how closely the performance of a heat exchanger in hydrogen can be predicted from the results of tests made in air, the data in the paper were applied to a commercial surface air cooler of the extended-surface type, manufactured by the author's company. This cooler was tested, using air and pure hydrogen and five runs were made with 8.22 per cent nitrogen in the hydrogen.

From the tests using air, the pressure drop and gas-side heat-transfer coefficient were determined. From these values, and the properties of hydrogen and nitrogen, the pressure drop and gas-side coefficient were estimated for pure hydrogen and a mixture of 91.78 per cent hydrogen and 8.22 per cent nitrogen by volume.

The pressure drop ΔP in inches of water is plotted against a modified Reynolds number in Fig. 5. The dotted lines show the estimated pressure drop based on the air tests, while the points show the values as obtained by actual test. It is apparent that

[4] Bibliography (8), p. 60.
[5] Bibliography (3), p. 214.

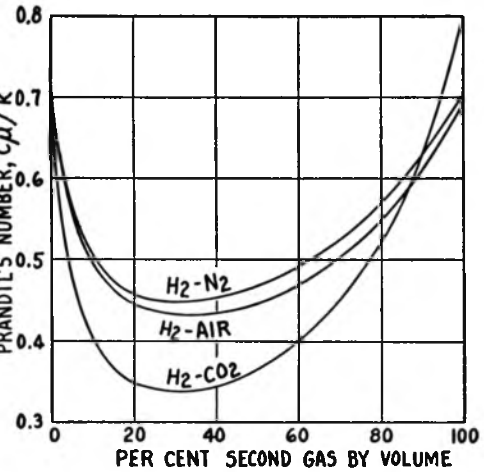


Fig. 4 Prandtl Number for Hydrogen Mixtures

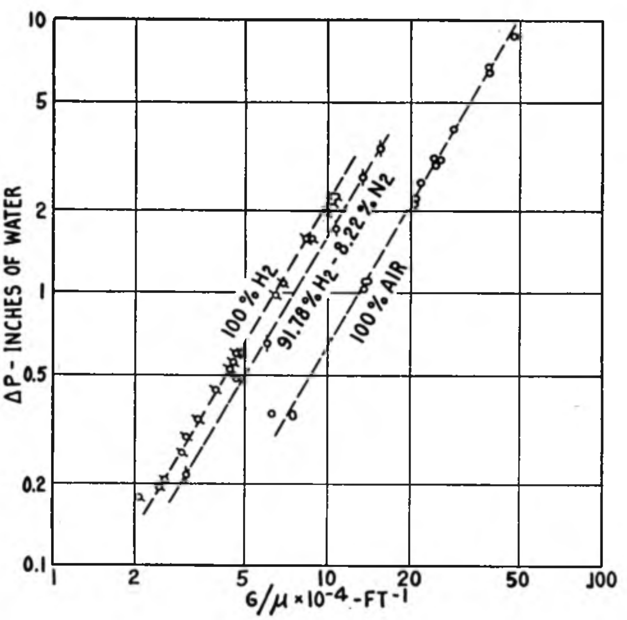


Fig. 5 Gas-Side Pressure Drop in Heat Exchanger; Comparison of Estimated and Test Values

the predicted values are well within the accuracy of the tests and it is possible to predict accurately the pressure drop of a heat exchanger when operating in a medium other than that in which it was tested.

In Fig. 6 is shown the local gas-side heat-transfer coefficient h plotted against a modified Reynolds number. Again, the dotted lines show the estimated values based on the air tests, while the points show the values obtained by actual test. Within the range of Reynolds' numbers covered by the air tests, the test values for the hydrogen lie close to the estimated curve. Below this range, the values depart from the estimated curve but are still within 10 per cent of the estimated value. This variation is due to the fact that the heat-transfer factor $j = \frac{h}{cG}\left(\frac{c\mu}{k}\right)^{2/3}$ is not a straight line when plotted against Reynolds' number but falls off slightly at the lower Reynolds number.

The points for the mixture of hydrogen and nitrogen are slightly below the estimated curve but, since only 4 points are shown, this cannot be conclusive. The 4 points shown are all within 10 per cent of the estimated value and the accuracy of the data on these particular runs was not very high.

Conclusion

The comparison of the estimated pressure drop and heat-transfer coefficient with the test values shows clearly that it is possible to predict the performance of a heat exchanger which is operating in a fluid other than that in which it was tested. Care must be exercised in extrapolating the available data but, within the range of the available test data, the estimated performance will be well within engineering accuracy.

It is the hope of the author that the information presented in this paper will further the application of hydrogen-cooled machines and will permit the extension of available test data on performance in air to cover the application of hydrogen and hydrogen mixtures.

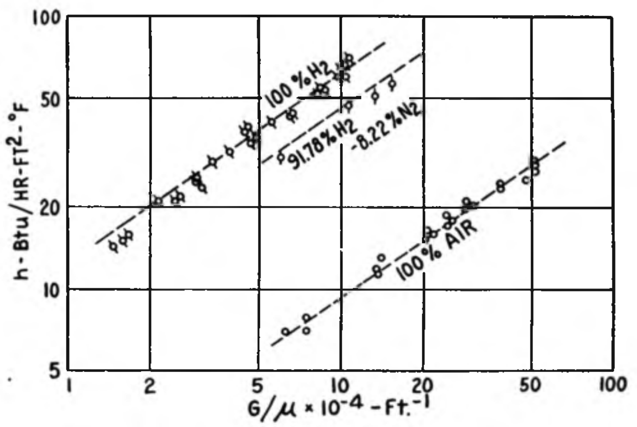


Fig. 6 Gas-Side Heat-Transfer Coefficient for Heat Exchanger; Comparison of Estimated and Test Values

Acknowledgment

Acknowledgment is made to Stanford Neal and S. L. Jameson, both of the General Electric Company, for assisting in the compilation of these data and for their cooperation and suggestions.

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Discussion

M. W. Beardsley.[6] Specific-heat data have been published since the references cited by the author, and there are now available equations[7] which cover substantially greater temperature ranges and have at least equal accuracy. For the temperature range of interest in this case, calculations can be considerably simplified by using the constant specific-heat values given in Table 2 of this discussion.

TABLE 2 CONSTANT SPECIFIC-HEAT VALUES

Gas	c_p, Btu per lb per deg F	Range, deg F	Maximum error, per cent
H_2	3.430	30–250	0.9
Air	0.243	0–250	0.5
N_2	0.249	0–250	0.3
CO_2	0.213	80–250	5.2
H_2-air	$\frac{6.91 + 0.085y}{2.016+26.81\ y}$	30–250 (all proportions)	< 1.0
H_2-N_2	$\frac{6.91 + 0.06y}{2.016+25.98y}$	30–250 (all proportions)	< 1.0
H_2-CO_2	$\frac{6.91 + 2.44y}{2.016+41.98y}$	30–250 (for $y \leqq 0.10$)	< 2.0

y = per cent by volume of gas other than H_2

The constant values shown for air and N_2 are actually more accurate over the cited range than the equations given in the paper. The value for H_2 is not as accurate as the equations above 100 F. The value for CO_2 is not recommended for accurate work, but the author's equation for CO_2 is over 3 per cent high for temperatures less than 80 F.

This information on specific heats is offered as data that can be more easily employed in calculations covering a limited temperature range.

W. M. D. Bryant.[8] The writer was gratified to observe that his gaseous molecular-heat data[9] had been of service in this research. The author has characterized these as "the most recent and reliable data." Neither designation is any longer strictly deserved, for several papers on the subject have appeared since 1933 and some have incorporated refinements of unquestioned value. For example, Murphy[10] has fitted a series of quadratic equations to some of the precision spectroscopic values later used by Dr. Heck. The loci of these equations are in some cases 2 to 3 per cent closer to the true values than those of the writer's equations. Also Sweigert and Beardsley[7] have derived some rather intricate equations which fit the spectroscopic curves with remarkable precision. However, it is doubtful that the use of these more recent data would alter the author's calculations to a significant degree.

H. W. Emmons.[11] The data used by the author in constructing his formulas appear to be the latest now available. As a convenient reminder, the valid temperature ranges for the various formulas should perhaps have been listed with the formulas rather than being buried in the text.

In general, the formulas chosen are those with some theoretical support, the constants being evaluated from test data. The exception, noted by the author, in the formulas for k is entirely justified from the point of view of available (and reliability of) present data. The Sutherland constant appears in the theoretical formula for k because of its direct appearance in μ and the theoretical prediction that $f = k/\mu c_v$ is constant. Since the Sutherland formula for μ fits the data for gases quite well (the author uses this form for air, nitrogen, and carbon dioxide), while the relation f = constant is only approximately true for actual gases at various temperatures, the thermal conductivity could not be expected to fit a formula with the same Sutherland constant.

The formulas for μ and k of binary mixtures adopted by the author do not agree with the latest theoretical work.[12] However, for the limited data now available, the empirical formulas with the empirical constants as given fit the data very well and are much easier to use than the theoretically more nearly correct equations.

The considerable variation of Prandtl number with composition is interesting and is nicely verified for practical purposes by the accuracy with which the heat-transfer performance for hydrogen-nitrogen mixtures can be predicted from the test data with air.

The experimental checks on the accuracy of calculations made, using the data presented, are excellent. The general value of the paper to those not familiar with the field would have been greatly increased if the author had devoted slightly more space to the dimensionless relations used in predicting performance of heat exchangers from air test data.

F. G. Keyes.[13] The paper is well developed as regards subject matter. A suggestion of possible interest relates to the theoretical work of Enskog who developed the kinetic theory of viscosity and heat conductivity on a very broad basis using the van der Waals molecular-field concept. This leads easily to a theory of mixtures which might find application.

G. W. Penney.[14] The effect of small percentages of heavier gases in hydrogen has been of interest ever since the cooling of machines by hydrogen was seriously considered. As plotted in Fig. 6 of the paper, the heat transfer for 91.78 per cent H_2 and 8.22 per cent H_2 appears poorer than that of pure hydrogen. However, in electric machines there are many cases where the velocity may tend to be more or less fixed by peripheral speeds. At a given linear velocity, a small percentage of a heavy gas usually improves the surface heat transfer. This fact made the effect of mixtures of considerable importance when hydrogen was first being considered. In studying this effect, tests were made at the Westinghouse Research Laboratories 12 to 15 years ago. Fig. 7 of this discussion gives the results of a test on a duct which showed a marked improvement in heat transfer when a small percentage of carbon dioxide is added to hydrogen.

The effect of a mixture of CO_2 in hydrogen on the surface heat transfer is shown when the fan speed was held constant in the duct system tested. This closely approximated the condition in many electric machines when the gas composition was varied in a given machine. Under these conditions the gas velocity was 6350 fpm for pure hydrogen and increased to 6460 fpm for 22 per cent CO_2 by volume, and to 7040 fpm for pure CO_2. The heat transfer plotted is the heat dissipated from the external surface of a cylindrical member $1^3/_{16}$ in. outside diam by 18 in. long and located concentrically inside a cylindrical duct of 2 in. inside diam. The

[6] Research Fellow, State Engineering Experiment Station, Georgia School of Technology, Atlanta, Ga.

[7] "Empirical Specific-Heat Equations Based Upon Spectroscopic Data," by R. L. Sweigert and M. W. Beardsley, Bulletin No. 2, State Engineering Experiment Station, Georgia School of Technology, Atlanta, Ga.

[8] Ammonia Department, Chemical Division, Experimental Station, E. I. du Pont de Nemours & Company, Wilmington, Del.

[9] Bibliography of paper (2).

[10] "The Temperature Variation of Some Thermodynamic Quantities," by G. M. Murphy, *Journal of Chemical Physics*, vol. 5, 1937, pp. 637–641.

[11] Assistant Professor, Mechanical Engineering, Towne Scientific School, University of Pennsylvania, Philadelphia, Pa. Jun. A.S.M.E.

[12] "The Mathematical Theory of Non-Uniform Gases," by S. Chapman and T. G. Cowling, Cambridge University Press, London, England, 1939, pp. 230 and 242.

[13] Professor, Research Laboratory of Physical Chemistry, Massachusetts Institute of Technology, Cambridge, Mass.

[14] Manager, Electro-Physics Department, Research Laboratories, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.

heat transfer plotted is the watts per square inch divided by the logarithmic-mean temperature difference.

In Fig. 7, the heat transfer is plotted for a gas velocity which varied only slightly. In order to apply such results more generally, tests for a range of velocities are needed, and it is desirable to plot the results on some dimensionless basis. McAdams in his book on heat transmission gives many examples of this. In these

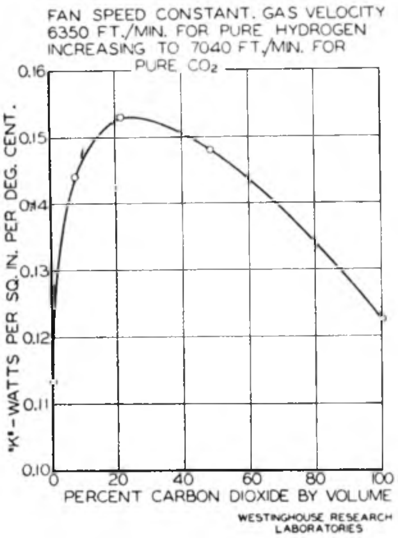

Fig. 7 Heat-Transfer Results of Test on Duct With Small Percentage of Carbon Dioxide Added to Hydrogen

examples the abscissa is usually related to Reynolds' number and the ordinate usually contains the Prandtl number ($C\mu/K$). By this means the three curves shown in Fig. 6 of the paper can be reduced to one common curve by the proper choice of coordinates. Such curves are valuable, but their usefulness is limited because a given curve applies actually only to geometrically similar surfaces. In the examples given by McAdams, the effect of the Prandlt number varies widely, depending upon the shape of the surface being cooled. For heat interchangers, many different shapes are used so that the experimental data reported are insufficient to cover the various designs being used.

The author has given a concise review of information which was previously scattered through the literature, but he has avoided the question which seems to the writer to be of major importance, namely, the application of these data to the multiplicity of shapes occurring in electric machines. When tests have been made on one particular shape, it is, of course, easy to secure good agreement with calculations on a geometrically similar surface. But it is difficult to predict the effect when a different geometrical shape is used.

It would be of value, if the author in his closure would include a dimensioned sketch of the heat-interchanger surface on which the tests of Fig. 6 were made. Each manufacturer usually has such data for his particular type of surface, but it is difficult to get sufficient data on different shapes.

R. L. Sweigert.[15] It is the writer's belief that the author has made a use of specific heats which does not seem quite necessary in dealing with hydrogen mixtures under the conditions under which those mixtures exist when used as a cooling medium.

The specific-heat equations used in the paper do not possess the accuracy stated, within the temperature range which the author gives. The error of the nitrogen and carbon-dioxide equations becomes very large at the higher temperature given in the paper. The error for the hydrogen equation is considerably smaller. The type of equation used is unsatisfactory for a wide temperature range. More recent accurate data and equations[7] on specific heats are available, the new-type equation being $a - \dfrac{b}{T} + \dfrac{f}{T^2}$ rather than $a + bT + fT^2$.

For ordinary temperature ranges, relatively close to atmospheric temperatures, older data may vary less than 3 per cent from the new, and the results from such data may come well within the limits of practical accuracy.

In the author's case, constant values of specific heat could have been used just as well, with accuracy equal to that of the equations used. Regardless of the possible inaccuracy of the results for higher temperatures from the specific-heat equations used, the author's results for the temperature range with which he was concerned were not materially affected, as is indicated by the agreement of calculated and experimental results.

[15] Professor of Mechanical Engineering, Director of Freshman Engineering, Georgia School of Technology, Atlanta, Ga. Mem. A.S.M.E.

Author's Closure

The principal criticism of all the discussers appears to be that the specific-heat data used may be 2 to 3 per cent at variance with the latest values. This fact was admitted by the author in his presentation of the paper but it must also be admitted that, at the present time, there is no marked agreement as to the specific heat even of air. In view of the fact that most heat-transfer calculations are excellent if they are within 5 per cent of the true value, this variation of 2 to 3 per cent is not of too great consequence.

In answer to Mr. Penny's question, as to the application of the presented heat-transfer data to "the multiplicity of shapes occurring in electrical machines," it was not the author's purpose to enter into the prediction of the heat-transfer coefficient for a multitude of shapes but rather to demonstrate the accuracy with which the heat-transfer coefficient may be predicted for any gas if the value in air is known.

Mr. Penny's conclusion that the heat transfer was poorer for the hydrogen mixture with 8 per cent N_2 than with pure hydrogen is true if we consider constant values of Reynolds' number. If a constant value of linear velocity or a constant value of weight velocity were used, the same would not be true. Assuming that $j = \dfrac{h}{cG}\left(\dfrac{c\mu}{k}\right)^{2/3} \propto \left(\dfrac{G}{\mu}\right)^{-0.25}$ then for values of constant G/μ, constant G and constant linear velocity (or G/ρ), we would obtain three different curves as shown by Fig. 8 of this closure. It may readily be seen that, under conditions of constant linear velocity, the coefficient rises and then falls off in much the same manner as in Mr. Penny's tests. If the Reynolds number or weight velocity are held constant, the heat-transfer coefficient falls steadily as the fraction of impurity rises.

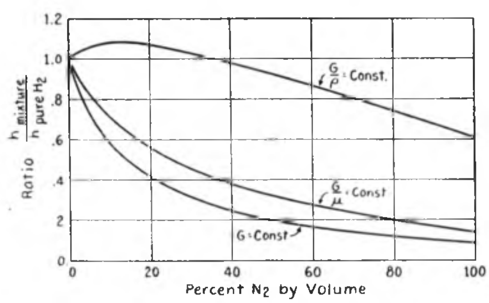

Fig. 8 Effect of Mixture Ratio on Heat-Transfer Coefficient

In electric machines, the velocity of the gas is nearly constant and is fixed by the machine speed. The heat-transfer coefficient may be improved by adding CO_2 or N_2 to the hydrogen. If this were the only result, this would be the proper procedure but it may be shown that adding nitrogen to 10 per cent of the total volume and maintaining constant velocity will give about 8 per cent increase in heat-transfer coefficient. It will also increase pressure drop by 75 per cent and the power required to circulate the gas by 75 per cent. In many cases, it is desirable to put in 8 per cent more heat-transfer surface in order to save 50 per cent of the power required; this fact must be borne in mind in all problems.

The author wishes to thank all the discussers for their helpful comments and criticisms and trusts that the data as set forth in the paper will prove helpful to those interested in hydrogen-cooled machines.

Experimental Drying

By D. L. COOPER and A. L. WOOD,[1] HALIFAX, N. S.

This paper deals with an investigation of moisture-regain values for salt-fish muscle which show a particularly rapid change in the range between 60 and 75 per cent relative humidity. The determination of experimental drying-rate curves in this range requires study, for much of the commercial drying as now carried out is done within this range. The author describes the system customarily used for this purpose. Dry- and wet-bulb regulators, employing thermionic relays capable of giving control of experimental air conditions in the order of 0.01 F dry and wet bulb, were used in the investigation. Drying-rate curves over a limited range using this method of control are presented.

THE experimental testing of materials, as a preliminary to commercial drier design and fabrication, is now recognized as essential in the development of such equipment. As a result, several types of laboratory equipment have been evolved for this purpose. In general, such equipment is suitable for the majority of requirements; it is only when an odd material is subjected to experiment that a more refined type of equipment is required. Necessary methods of securing satisfactory air flow are well established, but considerable improvement in control is now possible by the use of the newer types of equipment developed during the last decade.

The purpose of this paper is to describe certain refinements in experimental drier control but, in addition, an outline will be presented of the methods and limits of experimental testing as applied to the drying of salt fish, for the controls were developed to satisfy the requirements of this odd material.

Material to Be Dried

The material to be dried may be roughly considered as a biological structure, containing as formative matter 22 per cent salt, 56 per cent water, and 22 per cent protein. It may be further loosely defined as a saturated solution of common salt, given a definite form, and held by a substrate of protein, which was the original muscle of the fish. It has been shown that a more complex combination between salt and protein is formed which has an activity greater than the saturated salt solution itself but, since this could not be detected by equilibrium-moisture measurements, therefore, the first and more simple concept is sufficient for the needs of the drying engineer.

This protein-salt complex is highly hygroscopic, and severe shrinkage of the protein substrate complicates the problem. The formation of a tough, nearly impervious skin during drying under certain conditions is another factor which must be considered.

A further complication is introduced by the heat sensitivity of the system which, in common with all whitefish proteins, cannot be subjected to temperatures higher than approximately 72 F without breakdown. The temperature limit of salt-fish muscle depends to some extent on previous treatments but, in general, 76 F for short periods of time is the upper safe limit.

The problem may therefore be summarized somewhat as follows: To cause evaporation, into ambient air, of water from a saturated solution of salt, forming part of a system in which the remainder is a heat-sensitive, shrinking, skin-forming protein.

The theory of such removal will not be discussed but rather details will be given of the present state of experimentation, designed in a semiempirical manner to set up drying-rate curves without regard to theory, as a basis for drier design immediately required by the industry.

Little work of any value has been done on the drying of salt fish itself. Nor does the literature contain much information of interest on the evaporation from such solutions in protein base.

The complex nature of the material and the scarcity of information on like types made it imperative to disregard any previous work and lay down a program somewhat as follows:

1 Determination of the hygroscopy of the product, that is, determination of the regain values over all ranges likely to be encountered in practice.

2 To determine drying-rate curves for the material in experimental equipment.

3 To check drying rates thus obtained in commercial equipment.

4 To investigate the theory of evaporation from such systems.

So far results in connection with item 1 are complete; apparatus for item 2 has been devised and tested and a few results obtained. Work is proceeding on a commercial drier, designed and put in operation after the results of regain and drying rates were available. The theory is an interesting consideration which yet remains to be undertaken.

Equilibrium-Moisture Contents

A report of this work has been published previously (1).[2] Only sufficient detail will be given here to form a foundation for later discussion.

It is known that moisture regain in this class of material, particularly near the point of equilibrium, is very slow (2). Therefore, any method of determination, employing large samples, is likely to be slow. Unless conditions are controlled more accurately than is usually convenient, a slow method leaves a doubt as to whether the equilibrium recorded is a time equilibrium or one which represents the true equilibrium with vapor at a definite pressure (3). As will be shown later, this is particularly true in the case of salt fish in which the rate of change or regain is very sharp at high relative humidities.

The obvious method of making such measurements is by use of the McBain-Baker adsorption balance so arranged that the air may be removed from the measuring chamber.

In the experiments, the quartz spirals were turned to carry about 0.02 to 0.04 g of fish, and suspended in vacuo at constant temperature until equilibrium had been reached. Moisture at controlled pressure was supplied to these tubes, and an equilibrium read for each pressure of vapor admitted. Both the bath temperatures and vapor pressures could be varied to cover the range desired.

Results obtained in this apparatus are given in Fig. 1. The material was shown to be isohygrometric within the range and accuracy of experimentation, therefore, one curve expresses all the results.

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[1] Fisheries Research Bureau.

Contributed by the Process Industries Division and presented at the Spring Meeting, Worcester, Mass., May 1–3, 1940, of The American Society of Mechanical Engineers.

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

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Three things require mention: (*a*) The fish system becomes saturated at 75.3 per cent relative humidity, which is the theoretical value of saturated solutions of sodium chloride within the range of temperatures studied; (*b*) the exceedingly rapid increase in water regain at or near this saturation value; and (*c*) the downward and regain values do not agree over the whole range.

The supposition is that the protein system acts as a hygroscopic medium which causes uptake of water until a part of the salt is in free solution. The uptake of water will then be continued until the solution is diluted to give a vapor pressure in equilibrium with that of the surrounding medium. When humidities increased beyond 76 per cent, drops of liquid formed on the sample. These grew until they dropped to the bottom of the measuring tube.

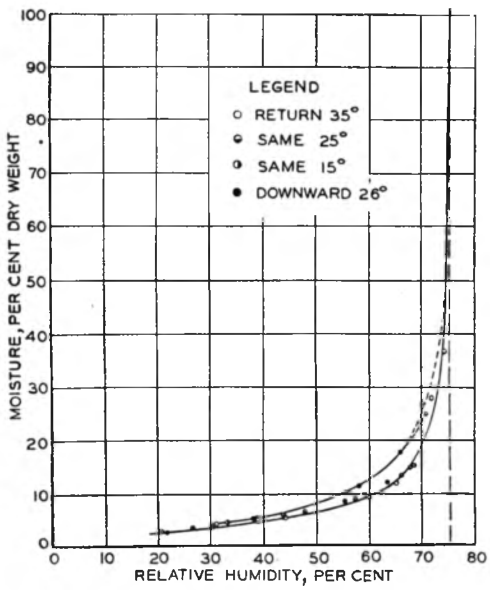

Fig. 1 Moisture-Regain Curve

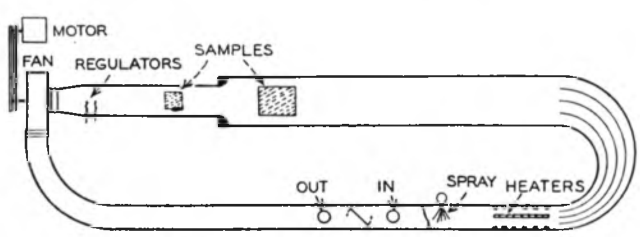

Fig. 2 Diagram of Experimental Tunnel

Much of the salt-fish drying in open-air flakes is carried out in relative humidities between 60 and 70 per cent. It is clear then from the curves for regain that, if this range is to be studied with the object of obtaining drying-rate data, the control of experimental drier conditions had to be somewhat better than that usually attempted.

Experimental Driers and Application of Controls

A most important consideration in the conduct of the experiments was the application of various types of regulating equipment suitable for the accurate control of conditions in the experimental driers. The systems to be described were all utilized in the experiments.

The drying equipment consisted of a conventional draw-through recirculating tunnel, Fig. 2, the design of which included the usual precautions for obtaining normal turbulent-flow distribution in those sections in which samples were suspended. No provision was made for dehumidifying air, low-humidity experiments having been carried out in winter months. Necessary air exchange to take care of evaporation and to maintain a humidity trend always lower than that desired was easily obtained by suitable adjustment of dampers.

Velocities were controlled at constant fan speeds by damper adjustments, or by varying fan speeds with adjustable drives. Temperatures were maintained with low-heat-capacity electric heaters, and humidities with a water-spray and manual damper control.

Control of Air Conditions in Small Tunnels

The difficulty usually experienced in obtaining accurate control in this type of equipment is caused by the lack of a sink or plenum which takes care of a lag or lead of the commercial type of control. This type will therefore be unsatisfactory except in circumstances in which the heat demand is constant and the supply accurately fixed to meet the demand. Otherwise, the lag together with the heat capacity of the heating elements will cause a hunting action, which becomes more serious if two dependent controls are used, as in the use of wet and dry controls for the regulation of temperatures and relative humidities. The modulating type constitutes an improvement in design. However, no commercial units of this type which were tried would give the required accuracy.

The best method of overcoming the difficulty is to use a combination of high-heat-output, low-heat-capacity heaters, which are operated by an on-off control of several times the required sensitivity.

The cause of failure in the commercial equipment tried appears to be the large currents required across the contacts of a thermoregulator. Even at low voltages and with careful contact design, cumbersome regulators easily disturbed by vibration were required, or regulators enclosed in an inert gas which are also fragile and subject to disturbance. Should this not be done, a change in the condition of the contact will cause excessive variations in the position of the control.

The solution of this difficulty is to use such low currents through the contacts that their resistance becomes negligible. This may be accomplished by several methods; as for example, the use of a very sensitive microrelay, which in turn operates a larger contactor carrying the heater or other load. The difficulties in the use of this system are caused by the sensitivity of the small relay which is usually also responsive to shock, dust, and other conditions. Several very sensitive and satisfactory relays are now available commercially but, as a rule, the second circuit which includes the main contactor must be carefully designed to avoid arcing at the contacts of the small relay. A better method is to use a thermionic tube as the sensitive relay or, if one of sufficient size can be purchased, as the only relay.

The author has tried both methods. The first always caused trouble, but the second has always been trouble-free. Several methods of operating thermionic tubes will be described, and details of others may be found in any standard text on vacuum tubes and their uses (4).

One major effect of systems of this type is to increase the response of common thermoregulators greatly. Thus, the bimetallic spiral regulator, usually sensitive to about $^1/_2$ deg in water and something over 2 deg in air of low velocity, when used with thermionic tubes, will give control of the order of 0.01 F in air.

System 1, shown in Fig. 3, is suitable where direct current is available and there is no objection to using a second relay to accommodate heavy loads. A suitable relay for use in this system is the mercury-in-glass type, tripped by a mechanism operated

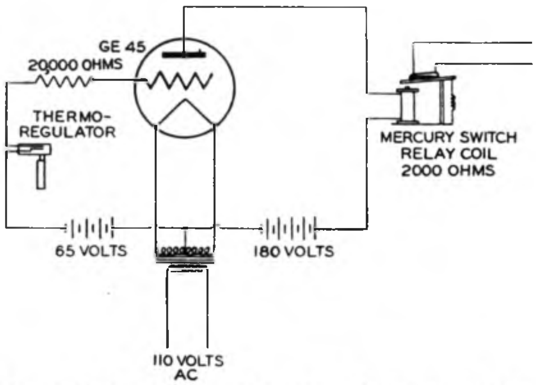

FIG. 3 DIRECT-CURRENT-OPERATED THERMIONIC RELAY

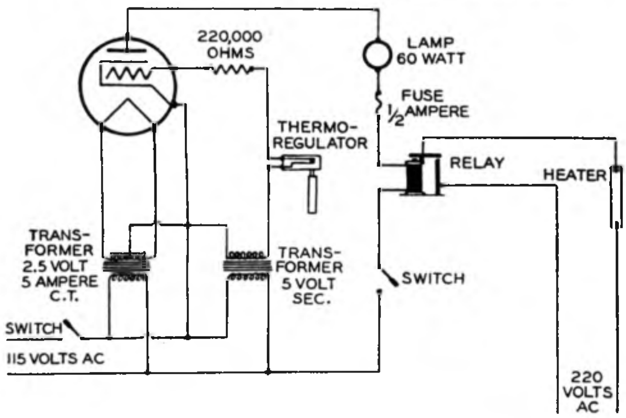

FIG. 4 ALTERNATING-CURRENT-OPERATED THERMIONIC RELAY

by a small magnet. The magnet may be wound to suit the characteristics of the tube used in the circuit.

System 2, shown in Fig. 4, employs the 3 or 4-electrode mercury-vapor-filled tube. One conventional method of use is indicated in Fig. 4, which employs an FG 97 thyratron. This tube is capable of handling small loads of the order of 300 w directly.

DRY-BULB CONTROL

It is clear that, if humidity is to be regulated by wet-wick thermoregulators, the accuracy of control of the dry bulb will affect the precision with which humidities may be maintained.

As previously mentioned, even for the most accurate control, the simple bimetallic type of regulator will be satisfactory if used in one of the systems already described. With suitable auxiliary equipment dry-bulb temperatures have been kept constant to about 0.01 F in this laboratory.

Accuracy of the type specified requires careful design of the auxiliary equipment. Relays, if used, must be of the quick-acting type and operate heaters of low heat capacity, the most suitable of which, for experimental use, is the open-wire electric type. These have very low capacity and no detectable lag. Furthermore, they can be easily adjusted to suit the load for cases where extreme accuracy is desired. A convenient method is to divide the heaters into two sections: (*a*) a variable-control bank, and (*b*) a variable-supply bank. The control bank operates from the thermoregulator through the tube and auxiliary relay (if needed), and the variable-supply bank is easily adjusted by hand, although it may be made automatic if required. The less the load variation, the closer these banks may be set, and the greater the accuracy of control. In practice it was found that, when using open-wire heaters, the supply setting could be varied about 50 per cent without affecting the accuracy materially.

Steam heaters may be used but only at the expense of some precision, since it is nearly, if not quite, impossible to reduce the heat capacity of such heaters to a negligible value for small experimental installations when working below steam temperatures. The thermionic type of control, operating some system of modulation for admission of steam, has been found to help somewhat, but great accuracy is difficult of attainment in spite of refinement of the thermoregulator circuit.

CONTROL OF RELATIVE HUMIDITY

Relative humidities may be controlled by two systems differing in fundamental arrangement: (*a*) a type in which the humidity-regulator setting is influenced by dry-bulb temperatures, and (*b*) a type in which it is not. The first generally employs the combination of wet- and dry-bulb regulators; the second uses a suitable isohygrometric material which is sensitive to changes in relative humidity. Either method has been found to be satisfactory, providing the precautions are such that the sensitivity of the regulator is several times the required amount, and the rate of vapor addition is adjusted approximately to the load.

The actual controls used in this laboratory have always required an addition of vapor to air, for all experiments have been carried out when natural psychrometric conditions permitted this system of control. Similar methods may, however, be applied when dehumidification equipment is available.

THE REGULATOR

Method 1. Almost any type of low-heat-capacity regulator may be used. In practice it has been found that the simple bimetallic-spiral type with the wet wick placed on the spiral gives satisfactory results, providing it is used to operate the control circuit of a vacuum-tube relay, which in turn operates to admit vapor or steam as required.

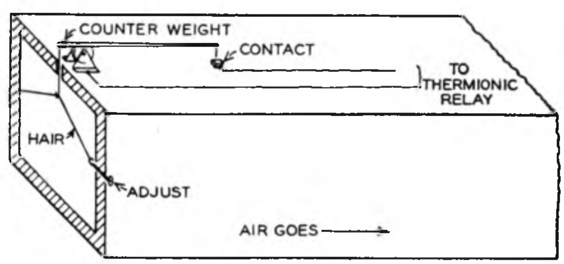

FIG. 5 HUMIDITY CONTROL

The familiar precautions of maintaining the wet-wick regulator in air moving at a velocity of 400 fpm or greater must be taken. In cases of experiments in which very low velocities are used, the regulator may be placed at the downstream side of a shaped nozzle, or fixed in a by-pass, the velocity in which is increased by an auxiliary fan.

Method 2. This method, employing the isohygrometric material, is equally as sensitive as the first, providing arrangements are satisfactory. The material may be either defatted human hair or white Russian horsehair, arranged as shown in Fig. 5. A fine wire, cemented to the long lever arm, ends in a contact which may be made as light as required, since contact resistances are negligible It is essential that the lever fulcrum be as frictionless as possible and that the lever be counterbalanced as shown to take nearly all the strain from the hair. The balance wheel of a cheap clock, mounted in the original frame, has been found to be satisfactory.

This system has the advantage in that it will maintain relative humidities constant, irrespective of changes in dry-bulb temperature. The hair must be reconditioned after each adjustment, and is, therefore, not suitable when the experiment calls for changes in relative humidities. Further, it loses sensitivity, and usually wanders, when the relative humidity goes above 70 or below 20 per cent.

The use of this system is particularly suitable for a predetermined dry-bulb cycle.

Introducing Vapor. Solenoid valves, operating to regulate the admission of steam or water vapor, are convenient for operation from the controls. Steam is most suitable where the quantity required is small and adds no appreciable heat load to the equipment.

Weight Recorders. It was necessary to continue drying operations for 12 hr or more on light samples, and small interruptions causing slight changes in air conditions affected the rates of loss of water. Therefore, it was desirable to incorporate some automatic method of recording weight changes.

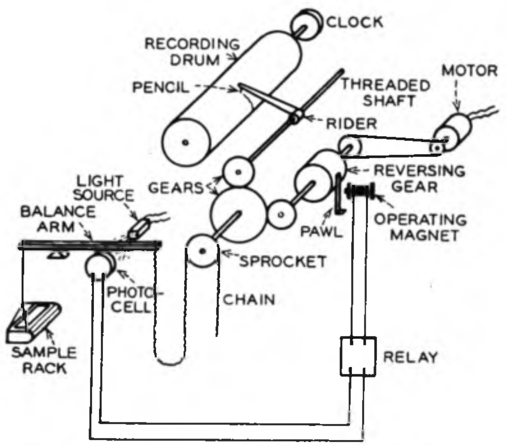

Fig. 6 Arrangement of Automatic Balance

Continuous weight recorders have been made in many types from the simple spring-and-stylo, to the various types of weight-dropping balances capable of great accuracy. The simple spring is generally not suitable for investigations on small samples, and the weight-dropping types are too expensive for the average laboratory.

A recording balance, Fig. 6, depending upon the action of a reversing gear, was devised (5) in 1935. It is the most versatile and suitable apparatus for recording loss in weight of a sample being dried.

This type of balance mechanism has the advantage in that it can be made from simple parts such as are supplied by any manufacturer of small gears. The reversing gears in use in this laboratory were made from standard Mecanno parts. These gears may be adapted for use with the finest chemical balances, or the common steelyard capable of measuring several tons. Since they maintain any weighing mechanism in dynamic balance, they increase its sensitivity markedly. A common steelyard loaded with several hundred pounds of rusty iron radiators will record differences in weight, caused by changing relative humidities, with consequent increase or decrease in the amount of absorption on the rusty iron (5).

The equipment consists of two parts: (*a*) the balance proper, and (*b*) the reversing gear with which is incorporated the recording drum. The reversing gear carries a chain, one end of which is attached to the balance, the other to a sprocket on the gear, which is connected to a worm carrying a pencil over the recording drum. The balance works as an automatic chainomatic balance and, if a simple ladder chain be used, the weight of the chain per unit length may be varied to suit the load expected.

The reversing gear is normally motor-turned to keep the sprocket moving forward. When the movement of the gear is interrupted by a pawl, acting on signal from the balance arm, the forward motion of the gear is stopped, the train of gears in the case coming into play and causing the chain sprocket to reverse.

At any position of equilibrium of the balance arm, the gear reverses about 30 times per min, keeping the balance in equilibrium about a point which is determined by the loss in weight of the sample. If the equilibrium is disturbed either by loss or gain in weight of the sample, the balance arm signals the change to the tripping pawl, which causes the gear to rotate in one direction until a new point of equilibrium is reached. This change is recorded on the drum actuated by clockwork and a continuous weight-change record is obtained.

The schematic diagram, Fig. 6, shows the essential actions of the recorder.

Figs. 7 and 8 show recordings of humidity control and weight

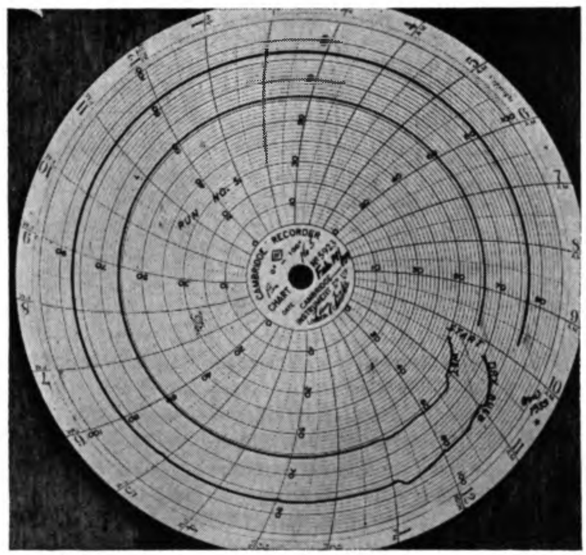

Fig. 7 Temperature Record of Run No. 8

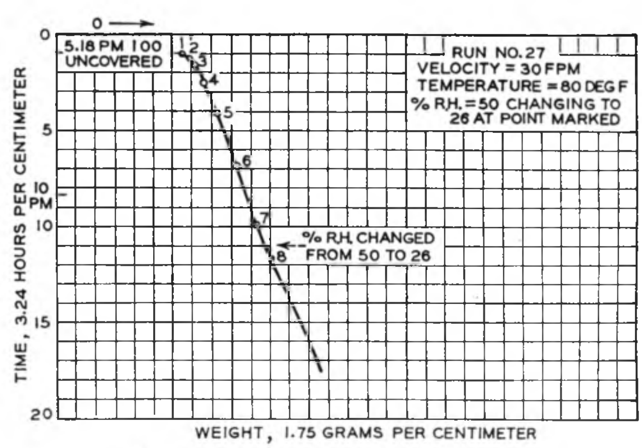

Fig. 8 Weight Record of Run No. 27

losses in experiments carried out at the laboratory with which the author is associated.

Fig. 7 was recorded by a Cambridge two-pen mercury-in-steel recording thermometer. The bulbs, being solid, were not particularly suitable for following rapid changes in air temperatures.

A mercury-in-glass Beckmann thermometer reading to 0.004 F, placed in the same plane as the recorders, showed variations of the order of 0.01 F. The control in this case was by wet and dry bimetallic spiral regulators acting through thyratron relays to activate (*a*) a self-atomizing spray for humidity control, and (*b*) a contactor which in turn operated electric strip heaters for dry-bulb control.

Fig. 8 is a copy of a typical weight curve taken directly from the graph drawn by the pencil of the automatic recorder. The recorder in this case was made of standard Mecanno parts. The drum was operated by a hand-wound clock. The balance was a common laboratory trip scale which had been in constant use about the laboratory for 4 years and was not in good condition. The weight of the sample and rack averaged 500 g; that of the sample itself about 35 g.

Experimental Results

Experimental drying curves were obtained in the usual manner by suspending the collodion-coated sample, fitted in a streamlined sample holder, from the end of the recording balance. The evaporation occurred from the top of the sample. All samples had a face area of 44 sq cm and were 7.1 mm thick.

TABLE 1 SUMMARY OF EXPERIMENTAL RESULTS

Run	Velocity, fpm	Relative humidity, per cent	Rate (avg), $cm^2/15\ hr \times 10^{-3}$
A	30	65	2.84
B	99	60	5.28
C	200	60	6.85
D	30	50	6.63
E	200	50	6.25
F	600	50	7.25

The results given in Table 1 are insufficient in number to permit conclusions of humidity, velocity, and temperature effects.

The curves shown in Fig. 9 give an idea of the nature of the drying process. The curves for all runs except *A* are similar in form to that of *E*.

One qualitative result of interest was noted as follows: When the average rate was less than approximately 3 to 4 $\times 10^{-3}$ the evaporation caused a heavy deposition of salt crystals on the surface of the sample. Above this range, the sample dried with a smooth nearly salt-free skin. It appears that this is a visual demonstration substantiating the theory of underskin evaporation in processes in which diffusion controls, and the drying potential is high. Measurable limits of undersurface evaporation

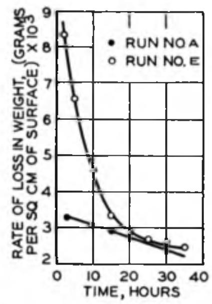

Fig. 9 Curves Indicating Nature of Drying Process

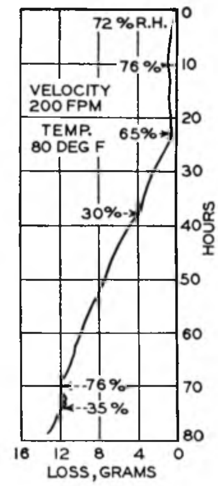

Fig. 10 (Right) Balance Record of a Run at Constant Dry Bulb and Varying Humidities

in many classes of materials should be susceptible of treatment by the use of salt solutions.

A curve of general interest taken from the recording balance is shown in Fig. 10. This was a miscellaneous run completed with the intention of investigating the correctness of the theory of regain as applied to drying. The theory is substantiated by this result.

In general the results obtained on experimental samples are not sufficient in number to enable analyses to be made. A new improved equipment is now in operation in which these results will be checked, and new ones obtained.

The results given herein were used to assist in the design of a commercial drier, now in operation for about 6 months.

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Heat Transfer to Falling-Water Films

By W. H. McADAMS,[1] T. B. DREW,[2] and G. S. BAYS, JR.[3]

Data are presented in this paper for heating water as it flows by gravity in turbulent motion down the inner walls of vertical copper pipes, ranging in height from 0.41 to 6.1 ft. Heat balances check within 5 per cent and the temperature of the inlet water ranges from 38 to 146 F. The data are adequately correlated by the equation: $h = 120\Gamma^{1/3}$, where Γ ranges from 600 to 15,000 lb of water per hr per ft of width of stream. Steam-side and over-all coefficients are given. The rather inadequate data for streamline flow by gravity over nearly horizontal pipes (trombones) are reviewed.

Nomenclature

THE nomenclature which is used throughout this paper is given in the following list:

- A = Area of heat-transfer surface, sq ft; A_i for inside, A_o for outside, A_{avg} for wall
- C = Specific heat of liquid, Btu per lb per deg F
- D = Diameter, ft; D_i for inside, D_o for outside
- g_L = Local acceleration due to gravity, approximately 4.18×10^8 ft per hr per hr
- h = Individual coefficient between tube surface and water layer, Btu per hr per sq ft per deg F; h_w for inside with turbulent flow, based on logarithmic-mean temperature difference; $h_{a.m.}$ for outside, with streamline flow, based on arithmetic-mean temperature difference
- h_{ST} = Individual coefficient between steam and outer wall, Btu per hr per sq ft per deg F, based on saturation temperature of steam and length mean temperature of wall on steam side
- k = Thermal conductivity, Btu per hr per sq ft per unit temperature gradient, deg F per ft
- L = Distance, ft; L for thickness of tube wall; L_H for height of heating surface; for vertical tubes, L_H is the height; for horizontal tubes, $L_H = \pi D_o/2$
- q = Rate of heat transfer, Btu per hr
- Re = Reynolds' number $= 4\Gamma/\mu$; Re_m is based on μ_m, Re_f on μ_f
- t = Temperature, deg F; t_1 for entering liquid; t_2 for exit; t_p for mean temperature of the surface of the pipe; t_{ST} for steam
- U_i = Over-all coefficient of heat transfer, Btu per hr per sq ft of inside surface per deg F logarithmic-mean over-all difference from steam to water
- w = Water rate, lb per hr leaving the tube
- x = Length of horizontal tube, ft
- Γ = Water rate from tube per unit width of stream, lb per hr per ft; for vertical, $\Gamma = w/\pi D_i$; for horizontal, $\Gamma = w/2x$
- (Δt) = Temperature difference, deg F; $(\Delta t)_o$, over-all from steam to water; $(\Delta t)_{ST}$, on the steam side; $(\Delta t)_p$, through the pipe wall; $(\Delta t)_i$, on the inside
- μ = Absolute viscosity, lb per ft per hr; $\mu_m = (\mu_1 + \mu_2)/2$; μ_f at film temperature; μ_a is taken at $(t_1 + t_2)/2$; μ_s is taken at t_p
- π = 3.1416
- ρ = Density, lb per cu ft

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Contributed by the Heat Transfer Professional Group and presented at the Annual Meeting, Philadelphia, Pa., December 4–8, 1939, of The American Society of Mechanical Engineers.

Introduction

The vertical shell-and-tube condenser, used widely in the refrigeration industry, is cooled by layers of water flowing by gravity down the inner walls of the tubes. Published information from several sources (1, 2, 3)[4] report over-all coefficients of heat transfer from condensing ammonia to water in such apparatus. No corresponding individual heat-transfer coefficients were available, and the work described in this paper was undertaken to fill this need.

Apparatus and Procedure

Three single-tube vertical falling-film heaters were constructed, each being heated by steam condensing on the outer wall of the tube. The three heaters have been described in an article (4), giving results for streamline flow of hydrocarbon oils. Table 1 summarizes the essential dimensions.

TABLE 1 NOTES ON VERTICAL APPARATUS

Heater no.	1	2	3
Heated length, L_H, ft	2.00	0.407	6.08
Inside diam of test section, in.	2.50	1.50	2.50
Outside diam of test section, in.	2.88	2.25	2.88
Metal for wall	Copper	Copper	Copper
Metal on steam side	Chrome plate	Chrome plate	Copper
Promoter on steam side	Oleic acid	Oleic acid	Benzyl mercaptan
Bakelite calming section, ft	0.50	0.75	0.75
Number of pipe-wall couples	10[a]	None[b]	10[c]
Jacket material	Steel	Pyrex glass	Copper-plated steel
Inside diam of jacket, in.	5.05	8	6.07
Filtered steam	Yes	No	Yes

[a] In heater No. 1 the five pairs of pipe-wall thermocouples were 1, 5.25, 12.25, 19.25, and 23.5 in. from the top of the heated section.
[b] In order to determine the average steam-side coefficient, a plot was made of $1/U_i$ vs. $1/\Gamma^{0.3}$, which gave h_{ST} of 10,000.
[c] In heater No. 3 the five pairs of pipe-wall couples were 2 7/16, 18 7/16, 34 7/16, 50 7/16, and 66 7/16 in. from the top of the heated pipe.

The saturation temperature of the steam was used in calculating the logarithmic-mean over-all temperature difference from steam to water $(\Delta t)_o$. The mean temperature of the outer surface of the tube was obtained by graphically averaging the ten temperatures over the length of the tube. The saturation temperature of the steam, less the mean temperature of the tube, gave the mean temperature difference $(\Delta t)_{ST}$ from steam to the outer wall. The mean temperature drop $(\Delta t)_p$ through the copper wall was calculated from q, k, L, and A_{avg}. The mean temperature difference $(\Delta t)_i$ between the inner wall and the cooling water was taken equal to $(\Delta t)_o - (\Delta t)_{ST} - (\Delta t)_p$. The heat balances checked within 5 per cent and the rate of heat transfer q was based on the water rate and the measured temperature

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

rise.[5] The following coefficients were calculated and are shown in the tables of data and results

$$U_i = \frac{q}{(A_i)(\Delta t)_o} = \frac{\text{Btu}}{\text{(hr)(sq ft inside)(deg F)}}$$

$$h_{ST} = \frac{q}{(A_o)(\Delta t)_{ST}} = \frac{\text{Btu}}{\text{(hr)(sq ft outside)(deg F)}}$$

$$h = \frac{q}{(A_i)(\Delta t)_i} = \frac{\text{Btu}}{\text{(hr)(sq ft inside)(deg F)}}$$

Tables 2, 3, and 4 show the data and results for one third of the runs, omitting duplicate runs which checked runs included in the tables.

Water-Side Coefficients in Vertical Tubes

It is clear from a study of the data that the water-side coefficient increases with increase in Γ, the water rate per unit width. Except for the runs[6] in which Re_f was less than 2100, the data for all three vertical heaters are plotted on logarithmic paper in Fig. 1 with h as ordinates versus Γ as abscissas. The average line AB corresponds to the equation

$$h = 120\Gamma^{1/3} \quad [1]$$

All the data of Fig. 1 lie between the two dotted lines laid off for deviations in h of ±18 per cent. For the longest tube, the runs with preheated water of series VII, shown by the triangles of Fig. 1, agree reasonably well with those where the water entered without preheat, series I–VI, shown by the circles. It was difficult to employ values of Γ greater than 22,000 due to a tendency of part of the water to arc across the weir and flow down the center of the tube, instead of flowing in a layer down the wall. It is possible that the insertion of a plug in the core of the water feed pipe would have prevented this. From the plot of h versus Γ for turbulent flow, it is concluded that heated length had no substantial effect upon h in the range from 0.41 to 6.1 ft.

Table 3 shows runs in which the temperature of the inlet water was varied substantially while Γ was held constant at various values. Examination shows that h increased somewhat with increase in the inlet temperature of the water. Table 3 shows Re_f based on μ_f and Re_m based on $\mu_m = (\mu_1 + \mu_2)/2$. The ratio $h/\Gamma^{1/3}$ was plotted against both μ_f and μ_m. In neither case was the per cent variation large for $h/\Gamma^{1/3}$, μ_f, and μ_m; the plot involving μ_m gave a somewhat better line than that with μ_f, and indicated

[5] Condensate was not measured for heater No. 2.

[6] Table 3, series L and part of series N.

TABLE 2[a] DATA AND RESULTS FOR NO. 1 HEATER

(L_H = 2 ft; t_{ST} = 227 F)

Run no.	Γ	t_1 deg F	t_2 deg F	$(\Delta t)_{ST}$ deg F	h_{ST}	U_i	h	$4\Gamma/\mu_m$	$h/\sqrt[3]{\Gamma}$
IV- 4[b]	2870	42.0	156.0	9.4	15000	1380	1670	5030	117
7	3060	42.0	155.0	9.2	16000	1420	1740	5260	120
10	3120	42.0	153.0	9.1	17000	1430	1720	5430	119
13	2600	43.0	162.0	9.2	15000	1350	1630	4650	118
16	2310	44.5	169.0	7.3	17000	1320	1560	4260	118
V- 1	2290	43.0	167.0	9.3	13000	1280	1550	4170	117
5	1830	44.5	180.0	7.8	14000	1240	1480	3450	121
VI- 2	2250	43.5	172.0	8.7	15000	1380	1660	4160	127
3	2050	44.5	177.5	8.0	15000	1340	1610	3820	127
VII- 3	2120	46.0	178.0	8.2	14000	1290	1560	3790	122
6	1930	46.0	180.0	8.2	14000	1290	1530	3700	123
VIII- 1	1760	46.5	181.5	7.3	14000	1220	1440	3400	120
4	1670	47.0	184.5	7.2	14000	1200	1430	3300	121
7	1530	46.5	186.0	7.3	13000	1130	1340	3020	116
IX- 1	1700	46.0	182.0	10.0	10000	1180	1440	3280	121
4	1610	49.0	184.0	8.5	11000	1180	1430	3200	122
7	1200	52.0	201.0	4.8	16000	1130	1320	2520	125
10	1470	51.0	192.0	6.5	14000	1190	1410	3060	125
13	3030	45.0	156.0	11.0	13000	1420	1760	5520	121
X- 2	3360	42.0	149.0	10.4	15000	1450	1790	5800	120
XI- 1	3600	42.0	144.0	13.1	12000	1450	1810	6120	118
3	4080	41.0	137.0	14.4	12000	1490	1870	6630	118
XIV- 1	3820	60.0	150.0	15.0	10000	1480	1910	5190	122
XV- 1	2120	62.0	173.0	12.9	8000	1180	1490	4820	116
3	2750	61.0	160.0	11.6	10000	1250	1540	6190	110
XVIII- 1[c]	3820	43.0	141.0	16.5	10000	1450	1860	6460	119
4	4070	42.0	137.0	14.4	12000	1460	1840	6850	116
7	4750	41.0	125.0	20.5	9000	1420	1880	7660	121
11	5620	40.5	117.0	22.3	8000	1480	1980	8830	111
13	2950	42.5	155.0	13.4	11000	1390	1940	5170	135
XXI- 1	3300	48.0	154.0	12.7	12000	1480	1860	6300	125
2	3320	60.0	154.0	13.6	10000	1370	1740	7160	117
5	3320	68.0	159.0	14.0	9000	1410	1820	7890	122
6	3360	79.0	166.0	11.3	11000	1480	1890	8880	126
7	3360	90.0	172.0	12.4	10000	1520	2000	9790	134
8	4490	60.0	142.0	14.9	11000	1520	1950	9450	118
9	2060	70.5	178.0	10.1	10000	1200	1490	5170	116
10	2060	77.5	180.0	10.4	9000	1190	1490	5500	116
11	2060	97.0	190.0	10.8	8000	1300	1690	6660	131
12	2060	88.5	184.5	10.1	9000	1220	1540	6170	120
13	2900	72.0	166.0	14.0	8000	1350	1750	7190	122
14	2550	82.5	175.0	11.5	9000	1300	1660	7080	122
XXII- 1	3350	58.0	153.0	16.6	9000	1470	1930	7500	127
3	4070	81.0	160.0	16.9	8000	1600	2170	10800	136
XXIV- 1[d]	3130	42.5	105.0			648			
4	3450	42.0	100.0			660			
7	4260	41.0	92.0			682			
10	4830	40.5	87.0			692			
12	2720	43.0	113.0			648			
13	2120	44.0	124.5			616			
15	1380	48.0	143.0			522			
17	2380	45.0	119.0			620			

[a] References (10, 12).
[b] Just before run IV-1, the inside of the tube was rubbed with a brush. Just prior to run III-1, seven test periods (10 days) before run IV-4 was made, the outside of the tube was swabbed with a cloth and treated with oleic acid.
[c] Just prior to run XVIII-1, the inside of the tube was rubbed with a cloth coated with fine emery powder, and several cubic centimeters of oleic acid were injected into the steam chest.
[d] Water flows upward, pipe running full.

TABLE 3[a,b] DATA AND RESULTS FOR NO. 2 HEATER

(L_H = 0.407 ft)

Run no.	Γ	t_1 deg C	t_2 deg C	t_{ST} deg C	U_i	h	Re_m	Re_f	$h/\Gamma^{1/3}$
F- 1[c]	4660	30.5	38.2	99.7	1360	1800	10500	15900	108
3	4660	26.5	34.7	99.7	1360	1800	9660	15800	108
5	4660	22.6	30.9	99.7	1300	1700	8890	15300	102
G- 1	3900	31.4	46.4	99.6	1350	1770	9050	13900	112
3	3900	26.4	35.8	99.6	1320	1720	8170	13400	110
5	3900	22.3	32.0	99.6	1210	1560	7550	12900	99
H- 1	3030	33.4	44.3	99.5	1350	1770	7400	11100	123
4	3030	27.3	38.8	99.5	1310	1710	6580	10600	118
6	3030	22.8	34.9	99.5	1250	1640	5900	10300	113
I- 1	2140	39.6	51.9	99.6	1200	1520	5940	8380	118
4	2140	32.6	46.2	99.6	1160	1480	5580	8390	115
8	2140	27.3	41.6	99.6	1120	1420	4310	7500	110
J- 1	1170	42.0	60.2	99.7	1090	1360	3500	4870	132
4	1170	34.6	54.6	99.7	1030	1280	3110	4570	124
8	1170	24.5	46.3	99.7	971	1190	2590	4280	116
K- 1[c]	5130	45.0	51.7	99.8	1630	2300	14900	20100	134
4	5130	39.3	46.4	99.8	1580	2250	13500	19200	131
7	5170	34.4	41.9	99.8	1540	2120	12500	18500	123
10	5170	27.9	35.8	99.8	1460	2000	11000	17700	116
13	5170	20.5	28.5	99.8	1350	1790	9350	16700	103
L- 1	420	21.0	55.3	99.9	574	640	935	1620[d]	85
2	420	22.5	60.6	99.9	676	770	985	1640[d]	103
3	420	23.9	62.9	99.9	710	810	1013	1680[d]	108
4	420	25.2	64.0	99.9	722	830	1040	1700[d]	111
5	420	27.0	65.9	99.9	752	870	1070	1710[d]	116
6	420	29.0	68.2	99.9	790	920	1130	1750[d]	123
7	420	31.0	69.6	99.9	798	930	1170	1770[d]	124
8	420	33.2	70.2	99.9	803	940	1200	1770[d]	125
9	402	36.0	71.3	99.9	750	865	1200	1770[d]	118
10	391	37.7	72.2	99.9	746	860	1190	1700[d]	117
11	391	39.1	72.7	99.9	738	850	1220	1700[d]	116
12	391	46.3	73.1	99.9	643	730	1330	1780[d]	100
N- 1	490	41.7	70.8	100.0	798	930	1560	2130	118
4	490	38.2	69.5	100.0	817	955	1490	2110	121
7	490	33.5	68.0	100.0	840	990	1400	2070[d]	125
10	469	29.8	66.8	100.0	820	960	1260	1940[d]	124
14	469	22.3	62.0	100.0	780	905	1110	1850[d]	117
P- 1[c]	650	49.7	71.3	100.0	872	1040	2250	2950	120
4	746	44.0	66.3	100.0	917	1100	2380	3210	121
7	735	38.3	63.4	100.0	918	1100	2140	3100	122
10	735	33.3	60.2	100.0	910	1090	2000	3000	121
14	735	24.4	54.0	100.0	875	1040	1710	2800	116
Q- 1	2650	44.3	54.1	99.9	1280	1670	7840	10700	120
4	2700	38.0	48.1	99.9	1190	1520	7140	10400	109
7	2680	33.0	44.1	99.9	1200	1530	6530	10000	110
10	2690	27.6	39.3	99.9	1170	1480	5920	9670	106
14	2690	20.9	32.9	99.9	1090	1360	5140	9120	98

[a] References (11, 12).
[b] The inside surface of the tube was swabbed with a wet cloth before each series of runs.
[c] Just prior to series E, K, and P, the outside surface of the tube was polished with rouge and treated with oleic acid.
[d] Runs in which Re_f was below 2100.

TABLE 4[a] DATA AND RESULTS FOR HEATER NO. 3

(L_H = 6.08 ft; D_i = 0.208 ft; D_o = 0.240 ft; A_i = 3.98 sq ft)

(Vertical copper tube, filtered commercial steam, promoted with benzyl mercaptan)

Run no.	$\Gamma = w/\pi D$	t_1 deg F	t_2 deg F	t_{ST} deg F	Δt_{ST} deg F	h_{ST}	U_i	h	$4\Gamma/\mu_a$ A.A.T. (e)	$4\Gamma/\mu_m$ A.A.V. (f)	$h/\Gamma^{1/3}$
I- 1[b]	5300	39.5	181.0	217.5	11.2	10000	1380	1770	14200	9240	101
II- 2	9500	39.7	166.8	217.6	14.6	12000	1940	2680	23700	16200	127
5	10700	39.0	162.5	217.6	19.3	10000	2040	3020	26100	18000	141
9	11400	38.7	153.7	214.7	17.7	11000	1980	2810	26600	18900	125
12	8840	38.7	171.7	216.5	12.3	14000	2000	2720	22600	15000	131
16	8370	38.7	171.0	216.9	19.4	8000	1850	2750	21400	14200	136
III- 1	10300	38.8	154.5	216.0	25.0	7000	1770	2720	23900	17000	125
4	9610	39.0	162.5	216.6	13.6	13000	1880	2530	23400	16100	119
7	7450	39.8	175.0	214.6	9.6	15000	1820	2350	19600	13000	120
10[c]	6490	39.7	187.0	221.1	9.0	15000	1780	2280	18000	11400	123
13	9650	38.6	161.8	218.7	13.2	13000	1830	2420	23400	16200	114
16	14000	37.5	145.5	219.6	16.3	13000	2070	2840	30800	22500	118
19	15800	38.3	137.0	222.0	19.5	12000	2010	2810	33300	25300	112
22	18800	38.0	130.0	223.5	16.7	15000	2130	2900	37900	29500	108
25	15400	38.5	139.0	222.0	18.5	12000	2000	2770	32800	24800	112
IV- 1	4900	39.5	192.0	223.2	11.2	10000	1430	1840	13900	8600	108
5	7170	39.7	180.6	224.4	14.7	10000	1690	2290	19300	12500	119
V- 1	7460	38.8	180.0	225.5	15.3	10000	1680	2330	19600	12500	120
4	7840	38.8	178.3	225.2	17.7	9000	1780	2520	20600	13500	127
6[d]	9050	38.5	173.3	225.2	10.7	16000	1910	2480	23200	15400	119
9	10100	38.5	167.0	225.2	13.7	14000	1930	2570	25200	17000	119
12	12100	38.5	159.8	225.2	13.2	16000	2090	2800	29000	20100	123
15	13600	38.5	152.1	225.2	13.8	16000	2090	2800	31200	22200	118
18	16400	38.5	141.7	225.2	16.2	15000	2200	3020	25300	26400	119
VI- 3	16400	38.6	133.7	222.4	18.6	12000	1960	2680	33900	26000	105
6	17500	38.5	131.0	222.9	19.7	12000	2020	2820	35500	27600	109
9	20000	38.5	123.8	222.9	19.4	13000	2030	2800	38800	31100	103
12	20900	38.5	120.0	222.4	18.4	13000	2000	2720	39500	30200	99
VII- 6	9610	145.7	195.5	220.7	6.7	10000	1730	2390	43000	41700	112
8	12900	111.0	178.8	220.7	10.7	12000	2030	2760	47100	44100	118
10	16500	88.6	158.5	220.7	13.7	12000	2040	2820	50500	46100	111
12	7150	125.7	195.0	220.7	6.4	11000	1540	1980	29600	27900	103
14	7110	101.0	190.5	220.7	8.0	11000	1610	2020	26200	23400	105
VII- 1	12500	87.3	173.0	220.7	9.7	16000	2120	2870	40700	35700	124
2	16500	79.4	157.1	220.7	13.4	14000	2170	3020	48000	42500	119
3	17800	81.9	155.3	220.7	13.2	14000	2180	3040	51800	46700	116
4	21600	71.6	141.9	220.7	15.2	14000	2260	3080	56100	50100	110
5	21700	73.0	143.3	220.7	16.1	14000	2300	3100	57000	51100	111
7	12800	111.0	178.7	220.7	9.7	13000	2010	2760	47000	44000	118
9	16300	89.3	158.7	220.7	13.2	12000	2010	2780	50100	45900	110
11	7170	125.2	195.0	220.7	6.9	10000	1550	2010	29700	28000	104
13	7000	101.7	189.7	220.7	8.0	11000	1540	2000	25800	23100	104

[a] References (9, 12).
[b] Two or three cubic centimeters of benzyl mercaptan were added just prior to run I.
[c] The steam filter was cleaned just prior to run III-10.
[d] Just prior to run V-6, the second addition of mercaptan was made.
[e] The viscosity μ_a corresponds to the arithmetic average of the terminal-water temperatures.
[f] The viscosity μ_m is the arithmetic mean of μ_1 and μ_2 at the inlet- and outlet-water temperatures.

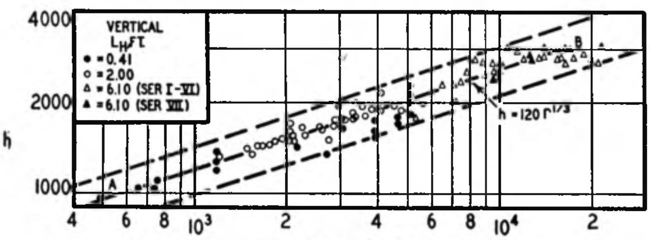

Fig. 1

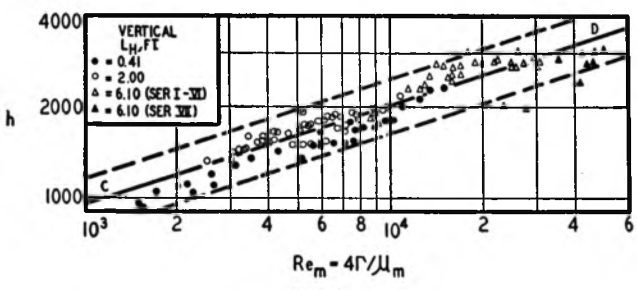

Fig. 2

FIGS. 1 AND 2 WATER-SIDE COEFFICIENTS FOR THREE VERTICAL FALLING-FILM HEATERS

(Re_f above 2100, plotted as h versus Γ Fig. 1, and as h versus Re_m Fig. 2; the correlation in Fig. 1 is preferred.)

that $h/\Gamma^{1/3}$ varied inversely as $\mu_m^{1/3}$. Consequently the data of Fig. 1 for all three heaters were plotted on logarithmic paper in Fig. 2 as h versus Re_m, and a line CD having a slope of one third was drawn, giving the relation

$$h = 95 Re_m^{1/3} \quad\text{[2]}$$

The bulk of the data lie between the two dotted lines which are laid off for deviations of ±20 per cent in h. Although the use of Re_m instead of Γ somewhat improved the correlation of the data for the shortest tube, the high-temperature runs for the longest tube, shown by triangles, show a deviation of nearly 30 per cent, whereas the same data in Fig. 1 did not deviate from the average line by more than 12 per cent. The correlation of Fig. 1 is preferred to that in Fig. 2.

OVER-ALL COEFFICIENTS

Heater No. 2 was also operated with an upflow of water so that the water filled the pipe. The corresponding over-all coefficients are shown in Fig. 3, compared with the data for downflow in layer form. In the range of Γ common to both types of

TABLE 5 RANGE OF HEAT-TRANSFER COEFFICIENTS AND OPERATING CONDITIONS

Heater no.	2	1	3
Heated length, ft.	0.407	2.0	6.08
Inlet water, deg F	59–123	41–97	38–146
Outlet water, deg F	83–164	117–201	120–196
Steam temp, deg F	211–212	227	215–225
Γ, lb water per hr per ft width	391–5170	1200–5620	4900–20900
$Re_m = 4\Gamma/\mu_m$	935–14900	2520–10800	8600–51000
h, on water side	640–2300	1320–2170	1770–3100
U_i, over-all coefficient	574–1630	1130–1600	1380–2300
h_{ST}, on steam side	(10000)	8000–17000	8000–16000

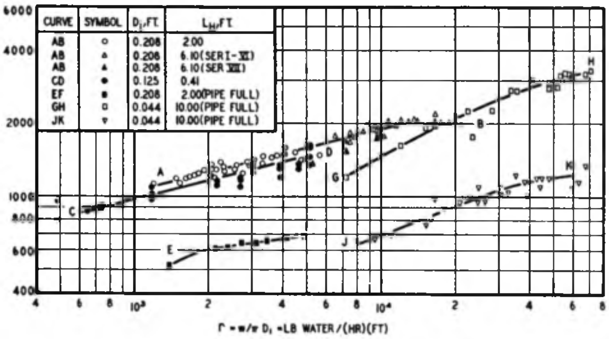

FIG. 3 OVER-ALL COEFFICIENTS FROM STEAM TO WATER FOR THREE VERTICAL FALLING-FILM HEATERS, COMPARED WITH DATA FOR TWO VERTICAL TUBES, RUNNING FULL

(The vertical falling-film heater AB gives over-all coefficients 2.2 times those EF for the same pipe running full; dropwise condensation on chrome plate, promoted with oleic acid, was used in both cases. For the 10-ft length of standard $^3/_8$-in. copper pipe running full [Ref. 6], the use of dropwise condensation on chrome plate, promoted with oleic acid GH gave over-all coefficients over twice those obtained JK without added promoter.* For curves GH and JK, a value of Γ of 10,000 corresponds to a velocity of 4.06 ft per sec.)

flow (Γ ranging from 1700 to 5000) the falling-film arrangement gave over-all coefficients 2.2 times those for upflow. This increase is due to the fact that the water velocity, for a given water rate, is higher for flow in film form.

STEAM-SIDE COEFFICIENTS

Abstracts of the steam-side coefficients for heaters Nos. 1 and 3 have appeared earlier (5, 6). Tables 2 and 4 show the results. The range of the values of h_{ST} for all three heaters is summarized in Table 5.

HORIZONTAL PIPES—WATER-SIDE COEFFICIENTS

Data (7, 8) for the gravity flow of water over nearly horizontal pipes lie in the region of streamline flow. For these runs it was assumed that half the water flows down each side of the pipe, and hence Γ was taken as equal to $w/2x$, where $w/2$ is the water rate for each stream, and x is the length of the pipe. As for vertical pipes, Re_m was taken as $4\Gamma/\mu$. Since the references (7, 8) did not give the water temperatures, it was necessary to base the coefficients for the horizontal pipes on the arithmetic-mean temperature difference, the type of mean customarily arbitrarily used for runs in streamline flow. Except at the low values of Γ, the resulting coefficients $h_{a.m.}$, are not far different from those which would have been obtained using the logarithmic-mean temperature difference.

For a given Γ, the data for streamline flow over horizontal pipes indicate that $h_{a.m.}$ is an inverse function of the height of the heating surface, $\pi D_o/2$. In the range of the data, for a given Γ, $h_{a.m.}$ varied approximately as $D_o^{-1/3}$; for a given D_o, $h_{a.m.}$ varied approximately as $\Gamma^{1/3}$, for values of Γ ranging from 150 to 400. As shown by Fig. 4, for values of Γ/D_o of 600 and more, the data for horizontal pipes may be approximated within ± 25 per cent by the equation

$$h_{a.m.} = 65\left(\frac{w}{2xD_o}\right)^{1/3} = 65\left(\frac{\Gamma}{D_o}\right)^{1/3} \qquad [3]$$

If the diameter were very large or Γ were very small, the water would be heated to substantially the temperature of the wall, and the final temperature difference, $t_p - t_2$ would approach zero. For the case of negligible evaporation, by definition

$$h_{a.m.} = \frac{(2\Gamma x)(C)(t_2 - t_1)}{\left(\pi D_o x\right)\left(t_p - \dfrac{t_1 + t_2}{2}\right)} \qquad [4]$$

The corresponding limit, for $t_p = t_2$ is

$$h_{a.m.} = \frac{4C}{\pi}\left(\frac{\Gamma}{D_o}\right) \qquad [5]$$

Equation [5] is plotted as the line AB on Fig. 4. If there were no evaporation, all points would have to lie to the right of AB approaching it asymptotically as Γ/D_o is decreased. At low water rates, the points on Fig. 4 are seen to lie to the left of AB, possibly indicating substantial evaporation.[7]

Data (7) for the three horizontal pipes fall within ± 25 per cent of the following semitheoretical equation based on stream-

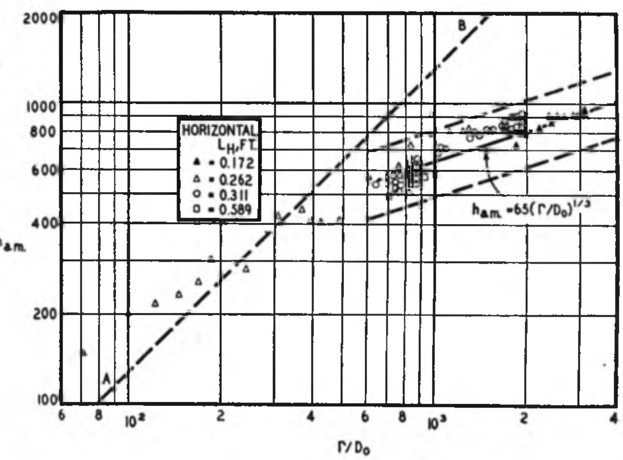

FIG. 4 WATER-SIDE COEFFICIENTS FOR HORIZONTAL (TROMBONE) HEATERS, PLOTTED AS $h_{a.m.}$ VERSUS (Γ/D_o)

(The dotted line AB represents the limiting case where the exit water is heated to wall temperature without evaporation, Equation [5]. In the range of Γ/D_o from 600 to 3000, the coefficients for the various sizes of pipe are roughly correlated by the equation $h_{a.m.} = 65\,(\Gamma/D_o)^{1/3}$.)

line flow of oils down the inner walls of the three vertical heaters (4)

$$\frac{h_{a.m.}}{k}\left(\frac{L_H k \mu_a{}^{1/3}}{C_p{}^{4/3} g_L{}^{2/3}}\right)^{1/3} = 0.67\left(\frac{\mu_a}{\mu_p}\right)^{0.25}\left(\frac{4\Gamma}{\mu_a}\right)^{1/9} \qquad [6]$$

This relation requires that $h_{a.m.}$ vary inversely as the one-third power of L_H, as shown in Fig. 4. The effect of viscosity on $h_{a.m.}$, predicted by Equation [6], is directly as $\mu_a^{0.028}/\mu_p^{0.25}$. Since in the water runs μ_p varied but little, no substantial effect of viscosity would be expected, and none was found (7, 8) for the water flowing over horizontal pipes. However, Equation [6] predicts that $h_{a.m.}$ varies directly as $\Gamma^{1/9}$, as was found to be the case for the oils, but the data for water on horizontal pipes show that $h_{a.m.}$ varies approximately as $\Gamma^{1/3}$, for Γ above 100. This increased percentage effect of Γ in the streamline region may be due to the presence of ripples in the water layer. The "horizontal" pipes (7, 8) were slightly inclined to facilitate flow of condensate, and the inclination would give an axial component to the water velocity. One (8) of those observers noted ripples in the water layer

* However, as explained in reference 6, "mixed condensation" was obtained in run JK due to a small reconcentration of promoter normally present in the boiler steam used. Data for film-type condensation are given as curve D-I of Fig. 1 of reference 6.

[7] The values of $h_{a.m.}$ were apparently based on the condensate rates. Although the rate of flow of effluent water was measured, it is not clear whether Γ was based on the rate of discharge of water or upon the mean of the rates of water feed and discharge. The rise in water temperature was not published, nor was the per cent evaporated given.

and found that the thickness of the water film at the side of the tube varied widely with position along the length.

Summary

Although predictions can be made most closely by using the plots of h versus Γ, and of $h_{a.m.}$ versus Γ/D_o, the following equations are recommended for approximate purposes:

For turbulent flow of water in layer form down vertical tubes, Γ ranging from 600–15,000 lb per hr per ft of stream width

$$h = 120\Gamma^{1/3}$$

with a deviation in h of ± 18 per cent.

For the streamline flow of water over nearly horizontal tubes, Γ/D_o ranging from 600 to 3000

$$h_{a.m.} = 65\left(\frac{\Gamma}{D_o}\right)^{1/3}$$

with a deviation in $h_{a.m.}$ of ± 25 per cent.

Acknowledgment

The authors wish to acknowledge the work of S. Baum (9), L. M. Blenderman (10), and S. T. Leavitt (11), who participated in the experimental work.

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6 "Dropwise Condensation of Steam on Vertical Tubes," by J. P. Fitzpatrick, S. Baum, and W. H. McAdams, Trans. American Institute of Chemical Engineers, vol. 35, 1939, pp. 97–107.

7 "A Horizontal Film-Type Cooler (Film Coefficients of Heat Transmissions)," by F. W. Adams, G. Broughton, and A. L. Conn, *Industrial and Engineering Chemistry*, vol. 28, 1936, pp. 537–541. The values of h are based on condensate rates, and the values of Γ are based on the rate of flow of effluent water. Standard 1, 2, and 4 in. iron-pipe-size copper pipes were used, and the tube was slightly inclined from the horizontal.

8 "Heat Transmission in Film-Type Coolers," by A. K. G. Thompson, Trans. Society of Chemical Industry, vol. 56, 1937, pp. 380T–384T. A tube of mild steel was used, having an outside diam of approximately 2 in.

9 "Heat Transfer in Falling-Film Heater," thesis by S Baum, The Massachusetts Institute of Technology, Cambridge, Mass., May, 1936.

10 "Heat-Transfer Coefficients in a Falling-Film Condenser," thesis by G. S. Bays, Jr., and L. M. Blenderman, The Massachusetts Institute of Technology, Cambridge, Mass., May, 1935.

11 "Overall Heat Transfer Coefficients From the Dropwise Condensation of Steam in a Falling-Film Condenser," thesis by S. T. Leavitt, The Massachusetts Institute of Technology, Cambridge, Mass., October, 1935.

12 "Heat Transfer in Falling-Film Heaters," thesis by G. S. Bays, Jr., The Massachusetts Institute of Technology, Cambridge, Mass., May. 1936.

Discussion

N. C. Ebaugh.[8] The writer would like the authors to answer the following questions:

1 What provision was made in the experimental apparatus for determining the quality or the superheat of the entering steam or, in other words, how was the enthalpy of the entering steam measured? The accuracy of the results has not been questioned since the heat balances were reported to check within 5 per cent. This is merely a request for additional information on the experimental procedure for problems of this type.

2 The over-all coefficient of heat transfer, when dropwise condensation was not promoted appears to be rather high when compared with other available data. Would the authors care to comment on the possible reasons for these high figures?

Authors' Closure

In reply to Professor Ebaugh's first question: High-pressure steam (at pressures usually ranging from 200 to 300 lb gage) from a well-drained line was reduced in pressure through a series of two regulators, giving superheated steam, the temperature and pressure of which were measured at a section just ahead of the jacket on the test section. The saturation temperature of the steam, corresponding to the pressure in the jacket, was used for computing temperature difference, and the enthalpy of the superheated steam was used in the heat balance.

The second question refers to the results (6) shown by curve *JK*, Fig. 3 of the paper. In the preprint, the subtitle to Fig. 3 failed to bring out the fact that a small amount of oleic acid was unavoidably present in the boiler steam used, although this was clearly stated (6). This defect in the subtitle of Fig. 3 was brought out in oral discussion, and the subtitle was revised at that time. The results of curve *GH* of Fig. 3 were obtained when using additional oleic acid above that normally present.

Curve *D-I* of Fig. 1 (6) gives results for a 6-ft × $^5/_8$-in-outside-diam vertical tube of Admiralty metal, operated where it was possible to employ atmospheric-pressure steam sufficiently pure to obtain film condensation, giving U_i of 500 at 6.3 fps.

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[8] Head Professor, Mechanical Engineering Department, University of Florida, Gainesville, Fla. Mem. A.S.M.E.

Temperature Distribution and Sources in the Conventional Railway Journal Box

By E. S. PEARCE,[1] INDIANAPOLIS, IND.

The author presents the results of an investigation on heat distribution in a conventional journal-box assembly, in the process, outlining the exact test procedure followed. He concludes that the control of heat sources is the first step in avoiding limiting effects of temperature rise in the assembly, and that the condition of the waste, the oil used, and the bearing are the basic factors in controlling these sources.

THE purpose of this paper is to set forth certain data illustrative of the distribution of heat generated in the conventional railway journal-box assembly. These data have been collected collaterally with an extended investigation of heat causes or sources and their control in journal-bearing operation on railroad rolling stock. In a sense, this is an uninterrupted continuation of research and development work described in a previous paper by the author.[2]

The same plant and equipment, as described in that paper, were used for these tests, the amplification of equipment being the application of some 53 thermocouples to a $5^1/_2 \times 10$-in. journal box and contained parts. Fig. 1 shows the box in place on the test plant. Fig. 2 shows diagrammatically the exact location of all the thermocouples and their numerical designation. Fig. 3 shows the method of attachment of thermocouples and their specific location in the bearing and wedge. Fig. 4 is a diagram of the internal arrangement of a thermocouple. Fig. 5 shows the location of the thermocouples in the bearing surface to obtain oil-film temperatures. These are thermocouples Nos. 7, 8, 9, 10, 11, and 12, Fig. 2.

Test Procedure

The preparation for these tests consisted of packing the journal box in the conventional manner with a good grade of wool packing, saturated with oil in the ratio of $3^1/_4$ lb of oil per lb of waste. The car oil used was of a standard brand, having a viscosity of 45 sec Saybolt at 210 F. The combined quantity of waste and oil weighed 8 lb, consisting of $1^7/_8$ lb of waste and $6^1/_8$ lb of oil. Conventional standard Association of American Railroads (A.A.R.) $5^1/_2 \times 10$-in. journal bearings, broached to insure a fixed width of crown, were used. All tests were conducted under a fixed total load of 20,000 lb, giving a unit loading of 890 lb per sq in. of projected actual bearing area.

Tests were run at operating speeds of 20, 40, and 60 mph, equivalent to 268, 536, and 804 ft per min journal surface speed, with rpm of 186, 372, and 558, respectively. A 7-hr continuous run was made at each respective speed, at which time all temperatures recorded were at their stabilized maximum. These temperatures are shown in Fig. 6. Atmospheric temperatures were maintained within practical limits at approximately those of high summer temperatures. Tests were duplicated under conditions of still air and under conditions of moving air, the velocity of the air being proportional to speed, and blowing against one side of the box in the conventional manner obtaining with the journal box under actual service.

For reference purposes, tests were run with the journal submerged in an oil bath to a depth of 1 in., indicated in Fig. 2 as "oil bath only," such tests being designated by this term. The same cycle was used as with the conventional waste pack. A sample comparison, in terms of temperature rise above atmospheric temperature, is shown in Fig. 6 for the temperature at each of the 53 thermocouple locations, as shown by Fig. 2. The temperatures shown in Fig. 6 represent in each case the maximum temperatures obtained after 7 hr continuous running at each respective speed under high atmospheric temperatures.

In general, the center of the journal has the highest temperature; the bearing has the next highest temperature; then, the oil film between the two; and then, the wedge on top of the bearing. The packing temperature is lower than the wedge and lower than the oil film. The temperature of the top of the box is lower than the wedge and the waste pack.

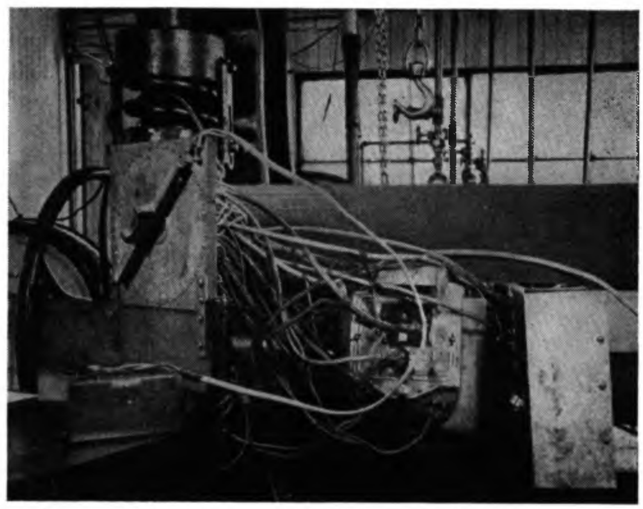

Fig. 1 Box in Place on Test Plant

The comparative temperatures, shown in Fig. 6, are not a measure of the relative proportion of friction losses between perfect bath lubrication and waste-pack lubrication. They have no significance in this respect at all. At 20 mph, the difference in friction loss between bath lubrication and the waste pack was 3.25 per cent; and at 40 mph it was 1.06 per cent, bath lubrication having the lowest loss. At 60 mph, the difference was 1.6 per cent, the waste pack having the lowest loss. No difference in friction could be expected between the two methods of lubrication, due to the fact that the bearing surfaces were receiving all the oil necessary for successful operation and all other conditions were the same.

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[1] President, Railway Service & Supply Corp. Mem. A.S.M.E.

[2] "Locomotive and Car Journal Lubrication," by E. S. Pearce, Trans. A.S.M.E., vol. 58, 1936, pp. 37–45.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, Philadelphia, Pa., December 4–8, 1939, of The American Society of Mechanical Engineers.

Discussion of this paper was accepted until January 10, 1940, and is published herewith directly following the paper.

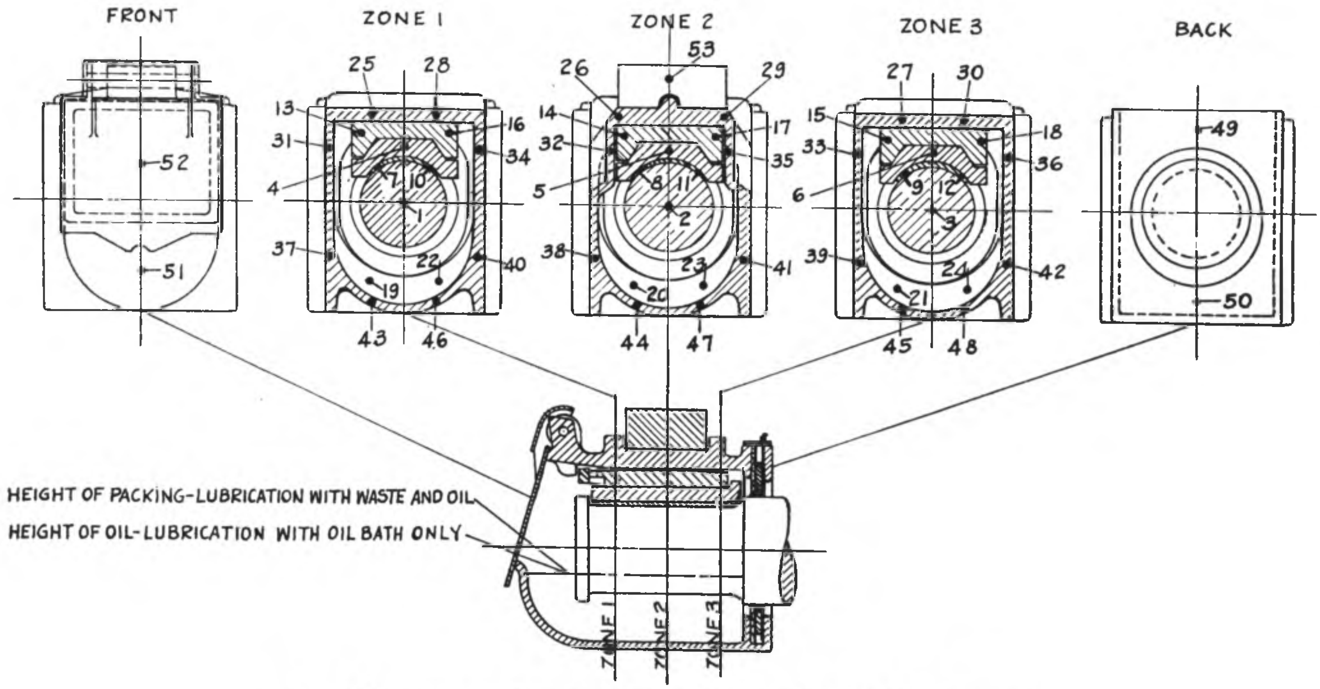

Fig. 2 Location of Thermocouples in Standard Journal-Box Assembly

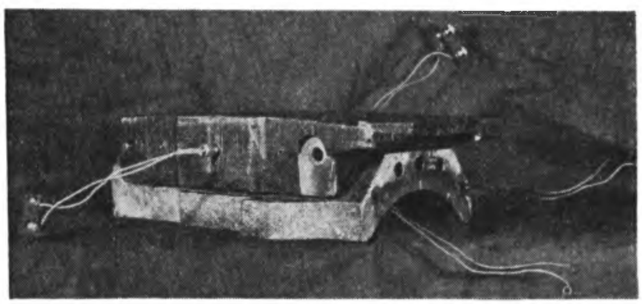

Fig. 3 Method of Attaching Thermocouples and Location in Bearing and Wedge

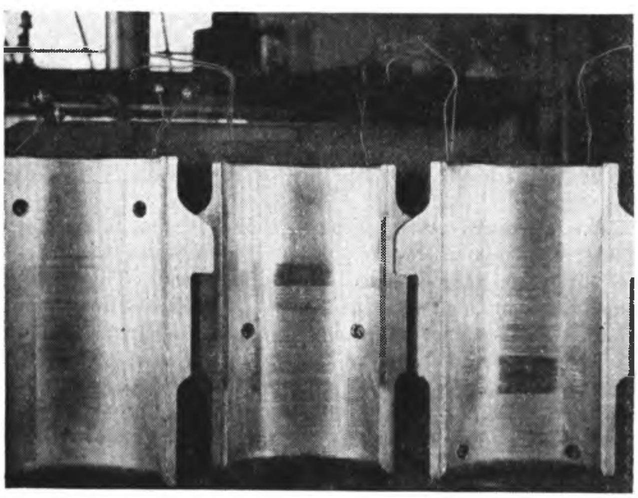

Fig. 5 Location of Thermocouples in Bearing Surface

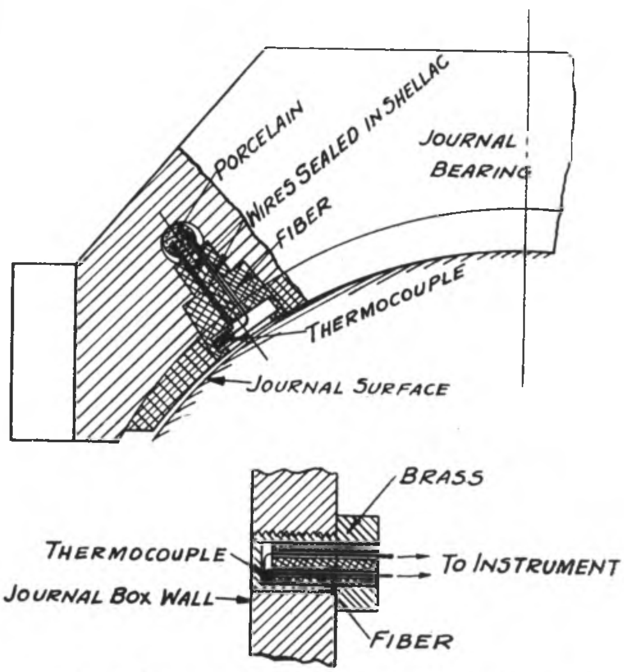

Fig. 4 Internal Arrangement of Thermocouple

(Top: Detail of application of thermocouple registering oil-film temperature. Bottom: Detail of application of thermocouple at various locations in journal box.)

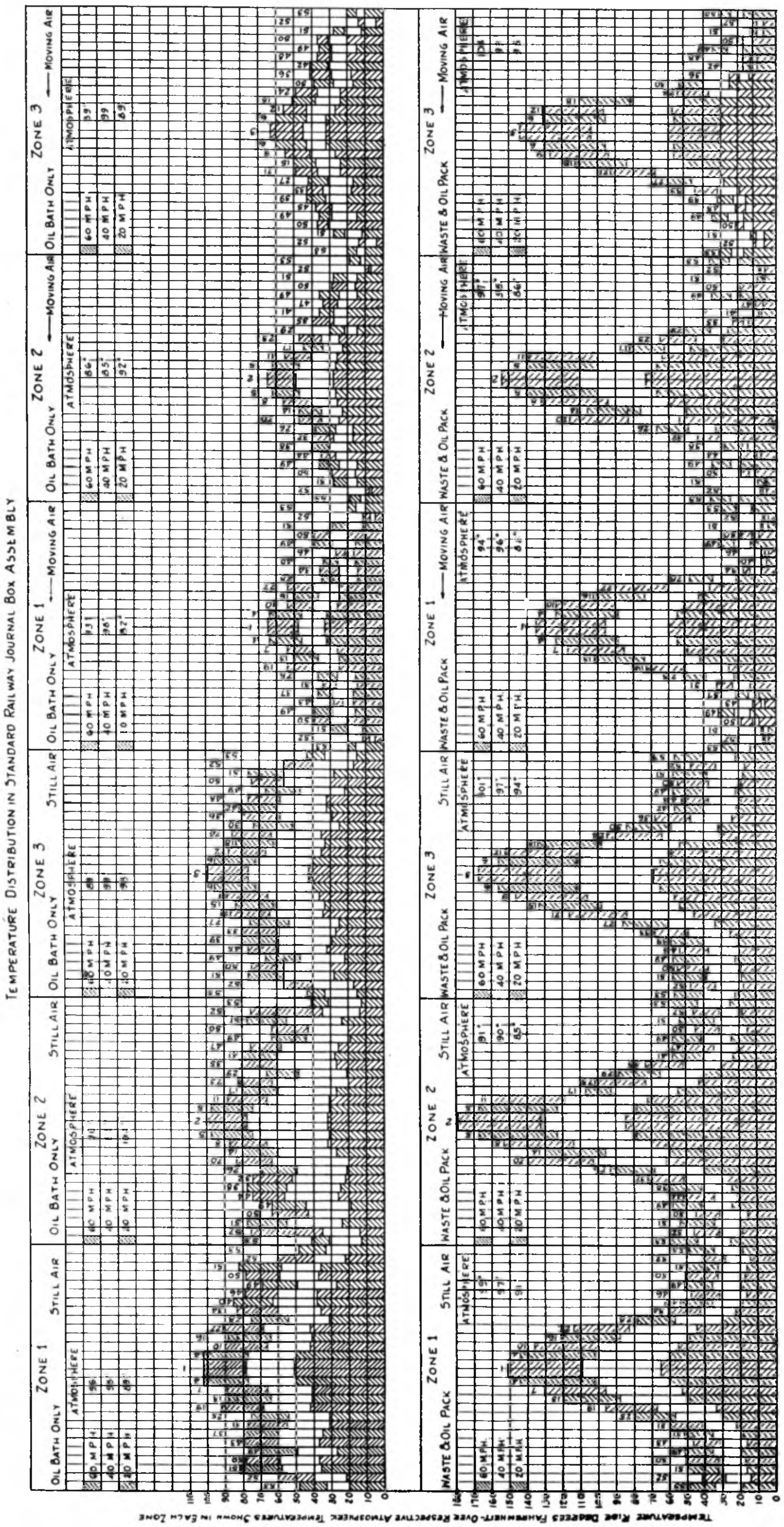

FIG. 6 TEMPERATURE DISTRIBUTION IN STANDARD RAILWAY JOURNAL-BOX ASSEMBLY

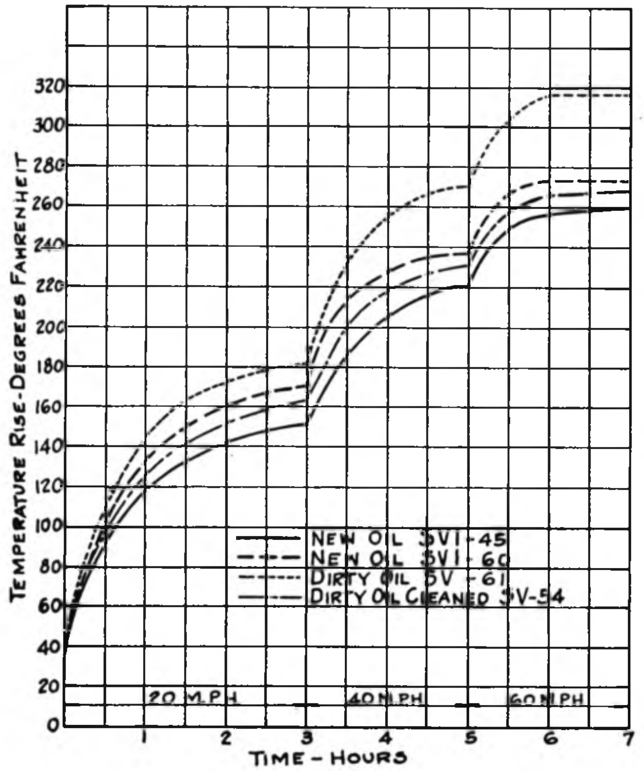

FIG. 7 TEMPERATURE AT CENTER OF JOURNAL UNDER NO LOAD AND WITHOUT BEARING

(Atmospheric temperature constant at 50 F; wool waste at 3.25 to 1 saturation; still air.)

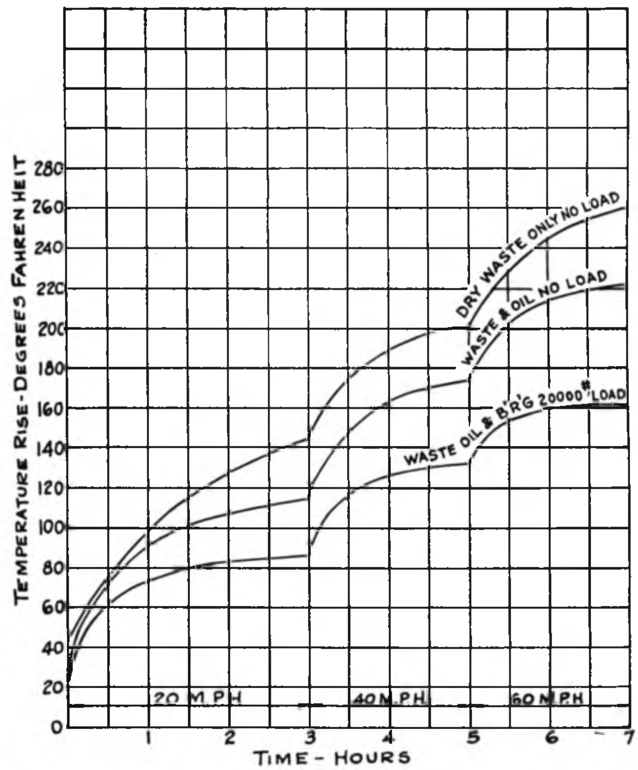

FIG. 8 TEMPERATURE AT CENTER OF JOURNAL UNDER VARIOUS CONDITIONS OF LOAD AND NO LOAD

(SVI-45 sec oil; atmospheric temperature constant at 50 F; cotton waste at 3.25 to 1 saturation; still air.)

Operation in still air and perfect flood lubrication have no practical application in the operation of railroad equipment. It is shown here only to reflect the insulating effect of the waste pack and the effect of air flow against the journal box. Railroad journals to operate at all are immediately subject to air movement. Flood, or bath lubrication, of railroad journals has many mechanical and operating limitations. Due to the fact that railroad journals operate under wide fluctuations of temperature, the insulating effect of waste might be considered an aid to lubrication in its influence on oil viscosity, which would otherwise be controlled by rapid temperature changes, as would exist with flood lubrication. The data reflecting the effect of flood lubrication in still air are to a degree representative of the operating conditions encountered in power-consuming or generating units of a stationary nature.

Effect of Viscosity and Cleanliness of Lubricating Oil on Journal-Box Temperatures

One common factor which influences the amount of heat generated in a journal-box assembly is the viscosity and cleanliness of the lubricating oil. To illustrate this, Fig. 7 reflects the temperature of the center of the journal under no load and without a bearing, comparing the temperature effect of new oils of 45 sec and 60 sec viscosity at 210 F, dirty oil as removed from service with a viscosity of 61 sec, and this same oil after cleaning with a viscosity of 54 sec at 210 F.

The effect of the bearing under load, as a source of increased journal temperature, is not a factor. It will be seen from Fig. 8 that the bearing serves to reduce the operating temperature of the journal by the fact that it aids materially in the dissipation of heat. This is because it provides the principal path for the transfer of the generated heat to the walls of the journal box. This obtains, however, under the condition of clean journal-box packing and with a bearing in proper mechanical condition. Dirty packing and/or a bearing in improper mechanical condition may be a limiting factor in operation by adding to the heat normally generated in excess above that possible of dissipation through the bearing. The modification of the conventional bearing to compensate further for the effects of temperature and to augment its heat-dissipating capacity has produced outstanding improvements in the actual service operation of bearings.

In Fig. 8 is also shown the temperature of the center of the journal as affected by clean dry waste, and with clean oil-saturated packing, without a bearing or load. Dry packing, or packing with a reduced oil content, approaches the temperature effect of the dry waste shown in proportion to the oil deficiency, which would be further augmented by increasing the dirt content of the packing, as reflected by the combined evidence of Figs. 7 and 8. In actual operation this condition is a source of heating, which is due to the effects of service, producing undersaturated dirty packing.

Conclusion

Obviously, the control of heat sources is the first step in avoiding the limiting effects of excessive temperature rise in the conventional journal-box assembly. The conditions of the waste, the oil, and the bearing are the basic factors in controlling temperature sources. There is no economic substitute for clean journal-box packing properly saturated. The proper conditioning of bearings at the time of initial installation, and their modification in design to compensate for the effects of temperature, increase the factor of dependable operation. The extensive reduction to practice of these fundamental principles by several railroads has amply demonstrated their value.

Discussion

G. B. Karelitz.[3] This comparison of temperatures recorded under several modes of operation and at various locations in the test bearing is rather interesting. In the waste-packed bearing, the temperatures of the journal and brass are considerably higher than those observed with the oil bath, but the wall temperatures are appreciably lower. In other words, the temperatures are more even in the case of the oil bath, while the temperature gradient from journal to wall is much steeper in the waste-packed bearing. This, no doubt, is caused by the oil thrown by the shaft from the bath onto the walls. The oil splash acts effectively in carrying heat from the journal and bearing to the walls. On the other hand, the heat generated in the oil film must travel in the waste-packed bearing by way of the brass, wedge, and bearing top to the bearing walls for final dissipation into the surrounding atmosphere by convection.

The temperature of the oil film was considerably higher in the case of the waste-packed bearing, the respective average temperature rise being 147 deg and 88 deg for waste packing and oil bath (60 mph, still air). The viscosity of the oil was, therefore, considerably higher with the oil bath, and the friction was correspondingly higher. This accounts for the higher wall temperature with the oil bath. The respective average wall-temperature rise was approximately 71 deg and 59 deg, indicating that the losses in the oil-bath bearing were some 20 per cent higher than in the waste-packed. This is at variance with the estimates given by the author. It would be of interest to know how the friction was measured, and to investigate the discrepancy.

The same general condition has developed in the experiments with moving air. However, in the absence of data on the character of draft and its velocity, no comparison can be drawn between the operation in still and in moving air.

Several years ago, the writer, while carrying on work with waste-packed bearings, also had occasion to measure the temperature at the center of the journal by means of a thermometer inserted in the journal. At that time, the temperature was found to be also the highest in the bearing. The heat transfer along the shaft is inappreciable because the heat conductivity of steel is rather low, and the temperature gradient along the shaft is small, due to the presence of a hot supporting bearing in the close neighborhood of the test journal. The center temperature of the journal is, therefore, fairly close to the average temperature of the oil film.

A further analysis of the data published by the author, with a full utilization of the test records and observations, can give a most interesting study in the heat transfer in a railway bearing. This would form a notable contribution to the general subject of temperature distribution in bearings.

C. B. Smith.[4] The conclusions in this paper merit the careful consideration of all interested in reducing the number of hot boxes occurring on passenger and freight trains.

The comparison of the high temperatures in charts, Figs. 6 and 8, indicates the significance of the journal bearing as a path for the dissipation of heat and directs attention to the proper relation to each other of the surfaces of the journal and of the bearing. Another chart might well have been included in the paper, showing a comparison of temperatures of the bearing with those at the center of the journal. This would illustrate more clearly the data shown in Figs. 6 and 8.

The heat-generating effect of dirty oil and the reduction in

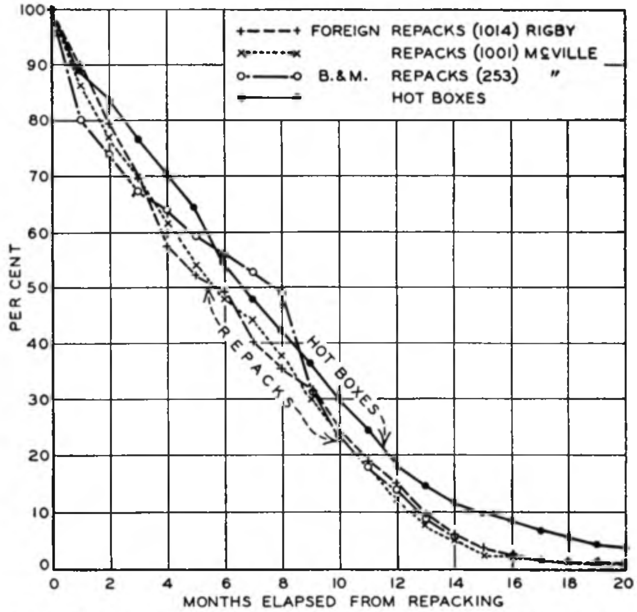

Fig. 9 Chart Showing Percentage of Hot Boxes in Relation to Repacks, Based on Months Elapsed From Date of Hot Box; Boston & Maine Railroad

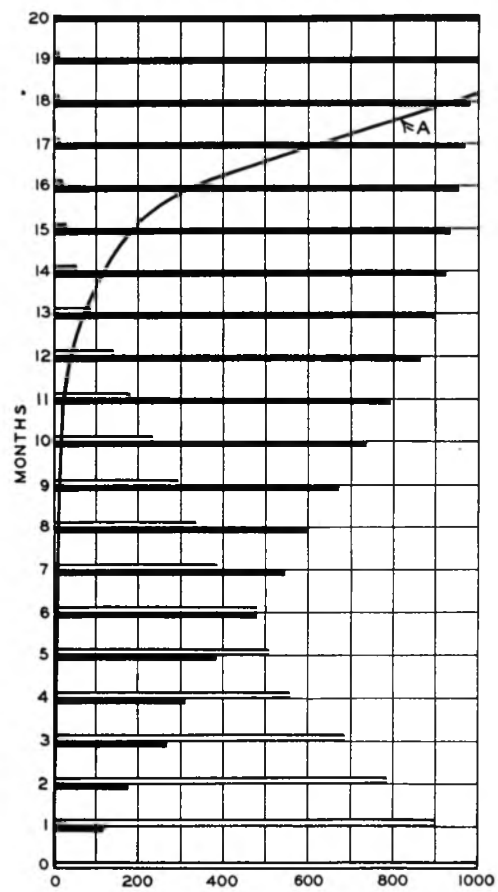

Fig. 10 Rate of Occurrence of Hot Boxes on Boston & Maine Railroad in Relation to Elapsed Time Between Repacks

(Length of open block line indicates cars that have packing dates not less than months shown. Length of solid black line indicates cars that ran hot with packing dates not more than months shown. Curve *A* indicates rate at which hot boxes accumulate as age average of packing in boxes becomes greater.)

[3] Professor of Mechanical Engineering, Columbia University, New York, N. Y. Mem. A.S.M.E.

[4] Engineer of Tests, Mechanical Engineering Office, Boston & Maine Railroad, No. Billerica, Mass.

temperatures attained after cleaning the dirty oil are impressively revealed in Fig. 7, even when no bearing and, hence, no load is involved.

The effect of dry or undersaturated packing in causing temperature increases ranges from 14 to 26 per cent, as shown in Fig. 8. This justifies free oiling of journal boxes when restoring cars to service which have been idle for a long period, especially in extremely hot weather. Fig. 6 clearly shows that the journal runs hotter than the bearing and emphasizes the maintenance of the best lubricating medium between the journal and the bearing, thus minimizing the heat transferred to the lead lining of the bearing.

The conclusion that the cleanliness of journal packing is economically important should impress those who favor the longer periods between repacking of journal boxes with clean, renovated packing waste.

The Boston & Maine Railroad has made some studies of the relation between the occurrences of hot boxes in freight-train service and the elapsed time between packing renewals. The hot boxes in freight trains on that railroad occur in cases where packing has been in service from 8 to 9 months. Figs. 9 and 10 of this discussion show that beyond 10 months elapsed time, the hot boxes occur at an increasing rate up to 15 months elapsed time and, thereafter, rapidly up to 18 months. A reduction of the repacking period to 9 or 10 months should show justifiable results.

Author's Closure

In the comment by Dr. Karelitz on frictional losses at 60 mph, comparing bath lubrication with waste-pack lubrication, since all conditions were the same on the entire test unit, except the absence of waste in one case and its presence in the other, total input in kilowatts per hour for each of the three 7-hour runs is a true basis of comparison for the determination of frictional losses with the following result:

60 Miles per hr, 7 hr, 20,000 lb load	Average kilowatts per hour — Test run — No. 1	No. 2	No. 3
With oil bath	2.59	2.58	2.61
With waste pack	2.59	2.74	2.70

Obviously the same amount of frictional heat must have been created in both cases. The fact that it was distributed differently to the walls of the box is a phenomenon which has no significance in establishing the quantity at the source.

The points brought out by Mr. Smith are particularly interesting, since they demonstrate that the results of the laboratory findings can be and are reproduced and their effect measured in terms of actual service. To illustrate more clearly, as Mr. Smith points out, the data shown in Figs. 6 and 8, Fig. 11 is included showing the comparison of temperatures of the bearing with those at the center of the journal, for the purpose of giving the effect of oils of approximately the same viscosity; but in the one case clean oil and in the other case dirty oil was used. These charts, taken particularly in connection with Figs. 9 and 10 shown by Mr. Smith, reflect in practical operation the effect of an accumulation of dirt in car-journal oil as it affects journal temperatures.

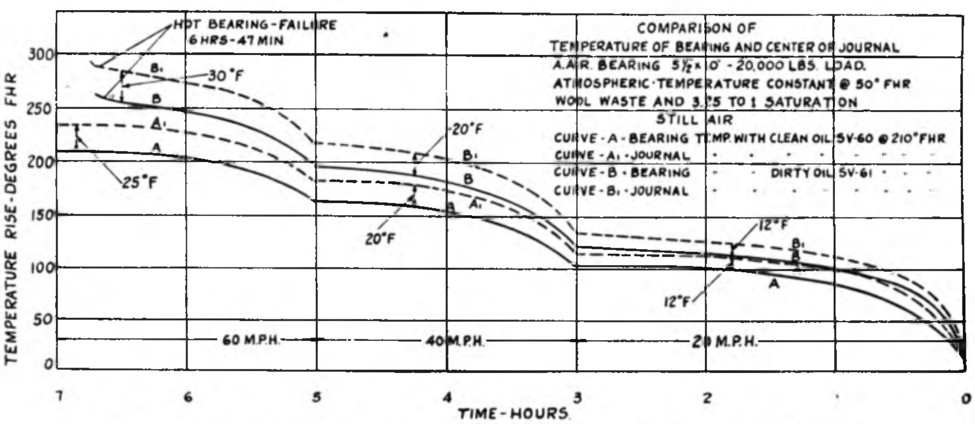

Fig. 11 Comparison of Temperature of Bearing and Center of Journal