THE BROWN BOVERI REVIEW

The art of engineers and the industrious hands of clever workmen create technical masterpieces.
The rotor of a gas turbine.

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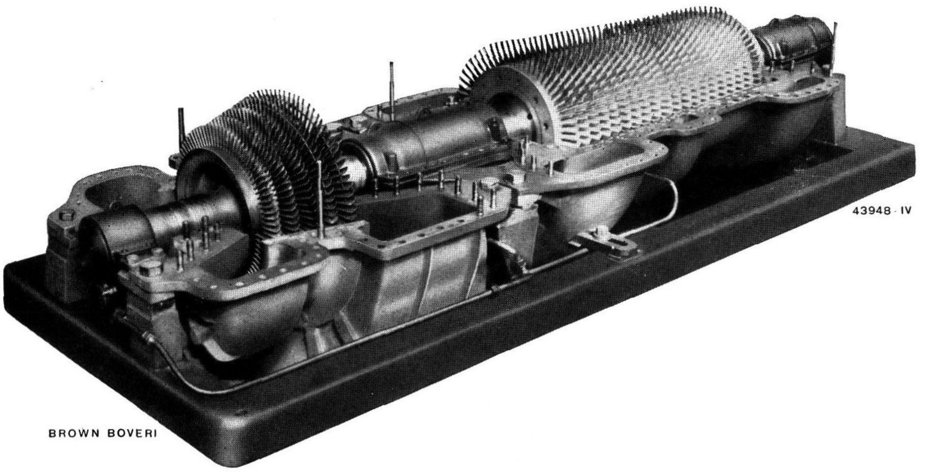
GAS TURBINES IN SERVICE

THE RESULT OF YEARS OF RESEARCH



The gas turbine as an independent power unit

is in its infancy. It will find an application wherever gas or liquid fuels are available and where importance is attached to low first costs, small space requirements, immediate readiness to take over load and where no water is available.

Our gas turbines have already found many practical applications:

as exhaust-gas turbines

for supercharging Diesel engines . . . . . (page 185)*
aeroplane engines . . . . (page 187)
producer-gas generators . . (page 206)
chemical processes . . . . (page 191)

as auxiliaries driven by hot gases

for pressure firing in steam generators (Velox) . (page 221)
in chemical apparatus . . . (page 191)
in metallurgical plants . . . (page 193)

as power units

to drive electric generators . . . . (page 192)
blast blowers in iron works . (page 240)
locomotives (page 236) and ships

* of this number.

It has long been the aim of engineers to create machines with rotating parts only. This object is attained with the gas turbine for the most important fields in which combustion engines are used.

THE BROWN BOVERI REVIEW

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PRESSURE-CHARGING, VELOX BOILER AND GAS TURBINE. A REVIEW OF THEIR ORIGIN AND DEVELOPMENT BY BROWN BOVERI.

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THE present special number treats of pressure-charging, of the Velox boiler, of the gas turbine, as well as of compressors such as are used for supercharging, for the Velox boiler and for the gas turbine. It gives an insight into a widely ramified domain of a most modern branch of engineering. The fact that we have contributed the greater part to these developments and that we were the first with many of them, fills us with just pride.

The first gas turbine and the first turbo-compressor.

We were not the first to take up the gas turbine, but a strange coincidence was responsible for our contributing to the realization of the first practical gas turbine (1, 5a, 6a).[1] In the year of 1906 we took up, in collaboration with Professor Rateau, the manufacture of turbo-compressors. The first machine to be built under this agreement was a compressor for a pressure of $4 \cdot 5$ kg/cm^2 abs, consisting of 25 wheels accommodated in three casings. It was in fact the first turbo-compressor (2) and was intended for a gas turbine being built at that time in Paris by Lemale and Armangaud. This gas turbine was provided with water injection to reduce the temperature of combustion, and the useful work was to be in the form of compressed air delivered to an existing distribution system.

The attainable compressor efficiencies were, however, at that time too low, as were also the gas temperatures which were limited by the relatively poor available heat resisting steels, so that but little success attended the result. Even so, the fact that it was possible to get the unit up to full speed represented a distinct advance over the first constant pressure gas turbine of Stolze (5b), notwithstanding the fact that the latter turbine already incorporated, as we shall see below, all essential features of the modern gas turbine with constant-pressure combustion.

More tangible was the success achieved in turbo-compressor manufacture following the first unit. This type of compressor found many applications first in the form of three-casing machines, but soon after as two-casing machines, especially for the production of compressed air in mining and steel plants (3).

The Holzwarth gas turbine.

Our interest in the gas turbine was next re-awakened by Holzwarth (5c, 6a). Rightly recognizing the diffi-

[1] The figures in ( ) refer to the literary index on pages 194/195.

culties in the gas turbine of obtaining a high compression and of controlling the high temperatures, Holzwarth endeavoured to overcome these by arranging for the pressure difference developing the useful work to be produced by the explosion of the mixture of fuel and air. Instead of one combustion chamber with continuous combustion, he proposed a number of combustion chambers, into which the mixture of fuel and air was admitted under the control of valves, exploded and discharged to the gas turbine wheel by means of a nozzle valve opening automatically under the pressure of the explosion. It was in February, 1909 that Holzwarth approached us, and on June 4th of the same year a contract agreement was signed. A first design of turbine which in the meantime had been constructed by the brothers Körting in Hannover, was taken over for test purposes and the decision was taken to build a 1000-H. P. trial machine. The designing of the turbine proper was done mainly by Holzwarth, we contributing from our experience in the manufacture of steam turbines that which could be usefully applied. Sustained periods of operation were achieved with this unit, but the fact that expectations as to the output and efficiency were not fulfilled did not, however, entirely surprise us, as we had in the meantime also studied thoroughly the theoretical side of the explosion gas turbine problem. Valuable experience had been gained, but as the matter made no substantial progress, our interest in the subject began to flag. In 1912 the further development of this turbine was taken over by the firm of Thyssen under Holzwarth's direction. We ourselves were seeking different ways of solving the problem.

The Humphrey gas pump.

Already at the time we were starting on the design of the Holzwarth turbine, rumours reached us from England of a new idea according to which the solution of the gas turbine problem was to be attained indirectly via a water turbine. These rumours referred to the Humphrey gas pump, in which a column of water was kept oscillating by the explosion of gas mixtures, the rising and falling surfaces of the two column ends acting as the pistons of a gas engine and of a pump (11). Apart from the valves, the device consisted only of simple tubular elements, and the efficiency was moreover very satisfactory (attaining under favourable conditions up to 28 %, referred to the water quantity delivered); it seemed, therefore, feasible to use this pump for delivering the motive fluid to water wheels. The only disadvantage was the relatively large dimensions. An attempt was, therefore, made to develop a two-stroke process instead of a four-stroke one. A small trial pump made in this manner operated well, but the same success did not attend a larger design.

Instead, however, of using the oscillating column as a water pump, it seemed feasible to use one of the end surfaces for air compression. Such a compressor promised high efficiencies and seemed, therefore, particularly suited for supplying air to gas turbines (5d). No practical unit was, however, attempted, owing to trials with the two-stroke process not having led to satisfactory results. Indeed, manufacturing work on the ordinary four-stroke type of pump, such as the two pumps for Argenta in Italy and the four powerful pumps of 400 H. P. each, which our licencees in Italy were building for Mex in Egypt, was brought to a sudden stop by the war in 1915.

All these labours were by no means completely wasted, for we learned much which was to be of use to us later. The principle of the oscillating water column for the operation of the gas turbine (the so-called "wet" gas turbine) was taken up again in substantially different form by others, among whom Prof. Stauber (6c) may be mentioned.

The gas turbine as second stage of a compound Diesel plant.

At the beginning of the world war of 1914, a fresh opportunity presented itself to us to return to the question of the gas turbine. In 1908 a patent was granted to Alfred Büchi of Winterthur, in which the idea of the compound combustion engine was again taken up, and wherein for the low-pressure part of the compression "a turbo-compressor, compressing as nearly as possible isothermally", and for residual expansion a gas turbine was suggested (10). The purpose of this compound machine was the reduction of the dimensions of Diesel engines, in order to make these machines lighter and cheaper and the pressures and temperatures more amenable.

Messrs. Sulzer Brothers in Winterthur, by whom Büchi was employed, decided towards the end of

1914 to make a trial, for which purpose a 20 H.P. four-stroke Diesel engine and a small Laval turbine were used. The pre-compression was effected by means of a reciprocating compressor (14). We participated in these tests, as it was proposed that whilst the further development of the reciprocating engine part should be carried out by Sulzer Brothers, we would take up that of the turbo-machine.

The results of these somewhat primitive trials were by no means negative. They were, however, not sufficiently convincing to be able to withstand the objections of prejudiced persons. In May, 1915, it was decided to drop the matter, as it was not considered possible to build the gas turbines and turbo-compressors so good that they would take the place of the reciprocating machine part of the unit. Doubts were also entertained about the high cylinder pressures which were inevitable with pre-compression. The main reason for the abandonment was, however, the fact left unspoken that it was not desired to hinder the already well advanced development of the two-stroke Diesel engine.

Thus, once again, had a new attempt of creating a gas turbine failed.

Incomparably more regular was the development of turbo-blowers and turbo-compressors. Already in 1912 it had become possible to accommodate the complete set of wheels of turbo-compressors for pressures up to 6 kg/cm^2 g in a single casing. Important for this development were the advances made in cooling. The arrangement of the coolers was responsible for the characteristic design and shape of the compressor. The cooling, but principally the improvement in the flow conditions in wheels and in the diffusors, which became the subject of a large amount of research, were responsible for a continuous rise of the efficiencies.

The latter were, however, still too low for the compressors intended for gas turbines. The period following the last world war, during which the cost of fuel rose continuously, called for more economical forms of power generation. High-pressure steam was much better able to cope with this requirement. The time of the gas turbine as a prime mover had not yet come, but there appeared another promising application, namely, as an accessory machine in connection with supercharging.

Supercharging and the exhaust gas turbine.

By "supercharging" we mean the delivery of a working fluid at an increased pressure for the purpose of increasing the output. Mainly turbo-blowers, generally called superchargers, are used for supercharging. They are best driven by means of a gas turbine, the operating medium for this gas turbine being the exhaust gases of the supercharged appliance. Among the appliances supercharged are combustion engines, but also furnaces and equipment employed in the chemical and metallurgical industries. As a rule, supercharging may be applied without altering the existing arrangements designed for operation without supercharging. To have recognized this fact and to have put it into practice is our claim, and it is thanks to the possibility in most cases of doing without any kind of external power supply, that the charging system enjoys its present-day popularity. Our first application of supercharging was to aeroplane engines. About the same time that in France Rateau, and in America Sherbondy proposed, in order to maintain the output of aeroplane engines at high altitudes, to supply the combustion air by blowers operated by exhaust gas turbines, one of our engineers made, in January 1917, the same proposal (4). It was, however, decided, with a view to speeding up manufacture, to drive the blower from the engine itself, or in case of large blowers by means of a separate engine. Already in 1918 large numbers of these compressors had been built by the Mannheim works of Brown Boveri (12).

The tests made with these superchargers both in reduced pressure chambers and at atmospheric pressure suggested the possibility of employing such compressors, driven by an exhaust gas turbine, for increasing the power of ordinary engines, but time passed before this fact was properly appreciated. We, however, went deeply into the question of this "supercharging", studying the problem from every point of view (13, 14). Finally, in September, 1923, a first practical application was realized. This supercharger with gas turbine drive was intended for raising the output of a new two-stroke Diesel engine of the Swiss Locomotive Works in Winterthur. The same set served later for our first real supercharging tests on a 500 H.P. four-stroke Diesel engine of the same firm, in which tests Alfred Büchi also participated. The collaboration of Brown Boveri, the Swiss Locomotive Works in Winterthur and Büchi led to the creation in October, 1926 of the Büchi Syndicate. The supercharging process was relatively quickly taken up by engine builders, this being due to the fact that we limited ourselves from the beginning to the supercharging of existing engine types. This meant, however, that the charging pressure had to be kept low, and hence that one had to be satisfied with the resulting moderate increase in power. As, however, the first engines to which the supercharging process was applied, comprised mainly conservatively rated land type engines or slow-running marine engines, even small charging pressures resulted often in substantial increases in output. The average charging pressure during the first years was of the order of $1 \cdot 3$ kg/cm^2 abs and the attainable increase in output ranged between 40 and 50 %. The gas turbine consisted of a single impulse wheel, the blower comprising mostly two radial wheels. Right from the start, special attention was given to the achievement of a high efficiency of the gas turbine and blower, as only so could the required excess air for scavenging the cylinder be ensured. Scavenging made it possible to improve the cylinder charge, to reduce the exhaust gas temperature and to cool the valves so that the supercharged machine, in spite of its higher heat rating, remained cooler than the non-supercharged one (20).

A substantial improvement of the scavenging principle was introduced by a patent of Büchi of the year 1925. It consisted of grouping together the exhaust of certain definite cylinders in common correspondingly dimensioned manifolds and leading them to separate nozzle groups of the gas turbine, thereby avoiding that the scavenging of one cylinder should be unfavourably affected by the gases just being discharged from other cylinders. The first stage of the supercharging of Diesel engines, comprising its introduction by the majority of European engine builders, fell within the years of 1926—1930. At the end of 1930 106 supercharging units had been delivered, the total output of the engines thus equipped attaining 200,000 H.P.

With the year 1930 started a new era for supercharging, brought about by the introduction of the so-called "exhaust impact process". In the case of engines operating on the exhaust impact process the charging set is brought as close as possible to the cylinders of the engine and limited numbers of cylinders are connected to a common nozzle sector by means of an exhaust pipe of minimum volume. Moreover, the nozzles are given a large section. In this manner it is possible to utilize the exhaust blast caused by the sudden opening of the exhaust valves for the purpose of developing power in the turbine and to attain a low pressure at the end of the discharge stroke. The exhaust impact raises the output of the turbine, and a lowering of the pressure facilitates scavenging. The exhaust impact process first realized by us in collaboration with the Maybach Motoren A.-G., Friedrichshafen (30), represents the extreme case of the previously mentioned proposal of Büchi, relating to the subdivision of the exhaust pipes for the purpose of preventing mutual interference during the exhaust and scavenging stroke. This process makes it possible to apply supercharging also to smaller engines, especially to the highly rated high-speed engines of average output such as are used for motor rail-coaches and submarines. Indeed, we build high-speed supercharging sets for motor lorry engines of only 50—100 H.P. To what extent the process of supercharging, now adopted by practically all the leading engine builders of the world, has grown in the last ten years, can be seen from the fact that at the end of 1940 the number of charging sets supplied attained 1500 and the total power exceeded $1 \cdot 6$ million H.P.

The blowers of the supercharging sets have only one wheel, which in certain cases attains very high circumferential speeds. In the case of the gas turbine the blading in particular has been considerably improved. The charging pressures obtainable with single-stage blowers lie between $1 \cdot 3$ and $1 \cdot 45$ kg/cm^2 abs and the increase in power obtained thereby is from 45—60 % of that of the machines without supercharging (29).

In the year 1940 we have developed a new application of the principle of engine charging, the particular importance of which is due to the present war conditions. Motor lorries must in ever increasing numbers be converted from petrol or Diesel oil to wood-gas. When operating with wood-gas, the motor loses 20—30 % of its former output. Supercharging enables this reduction to be more than made good. We, however, charge not only the engine, but also the producer, by inserting the charging compressor driven by the exhaust gases before the gas generator. The compressor has then only to compress cold air, smaller in quantity than the exhaust gases going to the gas turbine. With the supercharging it is easy to obtain an increase in power of about 30 % above that of the non-supercharged engine. The charging units are small machines, the diameter of the blower and of the gas turbine wheel in the case of an 80 H.P. lorry engine being of the order of 100 mm and running at speeds up to 45,000 r. p. m. The weight of the charging set is about 35 kg. With supercharging it is possible to gasify satisfactorily also soft woods, in particular pine wood (26).

The supercharging of aeroplane engines has ever since the first attempts in 1917 continuously received our attention. Considerable progress has been achieved, although up to the present only individual units have been built. The main difficulty of turbo-charging in the case of aeroplane engines lies, as is known, in the fact that the exhaust gases are very hot, so that it is not possible to ensure sufficient protection of the blades without the use of special means. We have investigated various possibilities of protecting them against heat. Among these may be mentioned the development of a blade in which direct contact of the hot gases with the metal of the blade is prevented by a thin layer of air distributed at the inlet edge over the whole width of the blade. Another method used with success is the supply to a part of the wheel periphery of cold air. More effective in our opinion, however, is the reduction in temperature of the exhaust achieved by liberal scavenging of the cylinder, as thereby the valves also are cooled. Scavenging is, however, only possible where fuel injection is used or a second air supply (for instance a second inlet valve) by-passing the carburettor, is provided.

Especially noteworthy are the high peripheral speeds of about 375 m/s of the compressor wheel now used in turbo-chargers for aeroplane engines, as well as the compression ratios attainable in a single stage of up to 3·5. This corresponds to maintaining constant the ground level output of the uncharged engine up to altitudes of nearly 8000 m.

The supercharging of two-stroke engines made relatively slow progress, the conditions being here considerably more difficult than in the case of the four-stroke engine, as the scavenging requires here a large additional pressure drop and entails a considerable reduction of the temperature of the exhaust gases. A patent of ours already applied in a number of cases consists in retaining the existing scavenging pump — which may consist either of a reciprocating compressor driven from the engine, or of a separately driven turbo- or cellular type compressor — and to supercharge this scavenging pump (32). The working conditions for the charging set are then, apart from the lower temperature of the exhaust gases, approximately the same as in the case of the four-stroke engine.

The reason why two-stroke supercharging still made slow progress lies mainly in the fact that the gain in output is considerably lower than in the case of the four-stroke engine. There are signs that development of the two-stroke cycle will in the future be more likely to take the form of a high-pressure compound engine with a gas turbine as a second stage, than that of a low-pressure supercharging arrangement; such a disposition being indeed the logical outcome of a development which also received our attention fore some time. From 1924 to 1927 we made many studies of arrangements of reciprocating engines with gas turbines on similar lines to a process proposed by Franke in 1912, in which the reciprocating engine drives a compressor and serves only for the compression of the working fluid, the latter being expanded in the gas turbine, delivering the useful power (5e). These studies were also extended to the floating piston engine, and a number of patents were taken out, all these being, however, left in abeyance, when almost at the same time two new problems presented themselves to us.

The first one was the re-tackling of the Holzwarth explosion gas turbine, the second a new steam generator, in many ways similar to the explosion gas turbine.

Back to the Holzwarth gas turbine.

Since his transfer to Müllheim on the Ruhr in 1912, Holzwarth had not been idle. The turbine which we had built was reconstructed by him at the Thyssen works and a series of further gas turbines were put in hand. Holzwarth succeeded also in gaining the interest of Professor Schüle and Stodola for his work. Both scientists carried out valuable investigations mainly in regard to problems concerning the theory of the turbine and the metals to be used; the last unit built, an oil-fired turbine, which operated quite satisfactorily, was made the subject of elaborate researches. The explosion turbine had in the meanwhile undoubtedly made such progress as to justify our again turning our attention to it. In August 1927 contact was re-established with Holzwarth, and in April 1928 a new agreement was concluded. As in the year 1909, it was proposed to build first a trial turbine, this time for an output of 2000 kW, using Diesel oil as fuel. The method of operation was to remain in principle the same as that upon which the first Holzwarth gas turbines were based, but the number of combustion chambers and the charging pressure of these were increased, so that considerably greater explosion pressures were to be expected. Because of the increased available pressure drop the expansion was to be distributed over two turbines, the first consisting of a two-row impulse wheel subjected to the varying explosion pressure, the second turbine, a multi-stage reaction turbine, absorbing the remaining drop with only slightly varying initial pressure. After one or two preliminary designs, for which the data were supplied by Holzwarth, one of the ten combustion chambers of the proposed machine was built, complete with all valves and the nozzle sector belonging thereto, and tested under service conditions.

During periods allowed for the tests, the final design of the turbine was to be settled and shop drawings were to be made. That this work was delayed and that one new design followed the other, came from manufacturing difficulties; that, however, the general lines and important details of the method of operation finally also underwent substantial changes was a result of the already mentioned explosion steam generator.

The explosion steam generator.

The incentive for the creation of this novel steam generator came undoubtedly from observations made on the explosion turbine, where much concern was caused by the magnitude of the heat losses to the cooling water, measured at certain places of the turbine. This could be explained only by the fact that the rates of heat transfer at the high gas pressures and velocities prevailing must be considerably greater than those given by the formulae hitherto employed. If this was indeed the case, then a tempting idea was to use these characteristics undesirable for the gas turbine in a device where it would be really advantageous, that is to say, in a boiler. Why then, not use the explosion process for the production of steam and limit the output of the gas turbine to a value just sufficient to drive the compressor of the combustion air?

The use of the explosion process and very high velocities of flow for purposes of steam generators was first proposed by us in 1927. In January 1928 followed the first patent applications. At the same time we built a small trial boiler, upon which the original test measurements of heat transfer were made. The subject seemed very promising, so that at the end of 1928 a ten-ton trial boiler for heavy oil, and soon thereafter a small test boiler for gas were put in hand. The latter operated for a while with blast furnace gas in Choindez, where the only blast furnace in Switzerland was at that time still in service (28, 15).

The method of operation of the explosion boiler distinguished itself from the working process of the original Holzwarth turbines in that a two-stage process was adopted. The upper stage comprised the final charging of the combustion chamber, the explosion, the delivery of heat and energy in the steam generator and gas turbine, the second stage comprised the scavenging and the precharging. The residual exhaust gases escaped through the economizer to atmosphere.

The Brown Boveri-Holzwarth gas turbine.

It was then reasoned that the above method of operation would also be suitable for a gas turbine, in that it resulted in a simplification of its design and an improvement of its efficiency. If, for instance, two combustion chambers were employed, the valve and nozzle sections being so dimensioned that the final charging and the discharge of the chambers would last as long as the precharging and the ejection of the residual exhaust gases, and if, moreover, the two combustion chambers were connected to the same nozzle sectors, the wheel should receive a practically uninterrupted stream. Further, in order to be able to operate the second turbine stage with a pressure varying as little as possible, notwithstanding the intermittency of flow characteristic of the explosion process, an equalizing container might be interposed between the two stages. Such a method of operation allows also of a combustion chamber arrangement favourable to liberation of the steam evaporated from the circulating water. Specially important is the reduction of the work of compression (34).

It was, therefore, decided in October 1930 to employ the two-stage process also for the trial gas turbine, and discarding the majority of the design projects made in the previous couple of years, work was accordingly started on the final workshop drawings. Manufacture began in 1931, and in July 1932 the first explosion trials were successfully effected in the turbine combustion chamber. The tests went on for a year. They were made with gas-oil and progressed somewhat slowly. It appeared then more advantageous to continue them with gas and to choose for the tests some place where gas was available and large amounts of energy could at all times be usefully absorbed without, however, necessitating absolute reliability. Holzwarth was successful in getting the gas turbine installed in the gas engine power plant of the August Thyssen steel works in Hamborn, where he reconstructed it for use with blast furnace gas, obtaining steadily improving operating results. When finally, in September 1937, a practically uninterrupted run of 470 hours was successfully concluded, the steel works placed an order for a 5000-kW turbine of the same design and operating on the same principle, but provided with twice the number of combustion chambers and wheels. The drawings were made by Holzwarth himself, and the manufacture was carried out entirely in Germany, mainly by our licencees. The turbine was put into service in the middle of 1940.

Whilst the exhaust gas turbine, which thanks to the unfailing perseverence of Holzwarth in overcoming all difficulties, made steady progress, the explosion boiler underwent already at the end of 1930 a fundamental alteration. Whereas the small gas-fired boiler operated entirely satisfactorily, it was found impossible to get the large ten-ton oil-fired outfit to function regularly. The main difficulty consisted in obtaining uniform explosions and attaining full pressure. The cause lay in the size of the combustion chamber which was over 1 m^3 in content; in the difficulty of ensuring uniform mixtures by means of a large number of mechanically atomizing injection nozzles, and perhaps principally in the incomplete scavenging, for which mainly combustion air and as little as possible scavenging air was to be used.

That the explosion process was persevered with so long was due to the fact that it was never expected to obtain with the constant-pressure process using high gas velocities, and with the same low compression requirements, either the very high heat transfer values which had already been measured with the former, or the extraordinarily high boiler efficiencies predicted by calculation. The future showed that this idea was a mistaken one.

The constant-pressure, or Velox steam generator.

At the end of 1930 the conversion of the test boiler to constant pressure was begun. The conversion was made in stages, a number of new problems having first to be solved. The most important of these was again the investigation of the heat transfer at high pressures and velocities and temperatures with steady flow. It should be recalled that the numerous researches which had been made up to that time on the transfer of heat, had been carried out only with relatively low gas velocities, that is, velocities far below the velocity of sound, and at relatively low temperatures, not associated with appreciable radiation effects. There had to be investigated also the effects of velocity and pressure on the combustion, in particular the influence so important for the creation of an intimate mixture and for the combustion, of the expenditure of large amounts of energy for creating intensive turbulence at the burner and at the air inlet, as well as the admissible loading of the combustion chamber per unit volume and per unit section. To this should be added the design of new type of burner for heavy oils and combustible gases. The furnace loadings of 6—10 million kcal/h and m^3 which were found to be possible with pressure-firing, resulted in very small combustion chambers. It was found also that even for large outputs it was advisable to use only a single injection nozzle, which, due to the small combustion chamber diameter, could only be at a short distance from the walls. The problem was, therefore, to develop a burner which could atomize up to 8 t/h of oil in a single nozzle, ensure good mixture with the combustion air, which could be regulated down to low loads, and which in order to prevent formation of coke, would not spray fuel on the combustion chamber walls at a distance which even in the largest boilers is less than 1 m.

It would not have been logical to reduce by the use of pressure firing and high gas velocities the dimensions of the combustion chambers and the heating surfaces to very small values and to retain the large steam drums hitherto employed. The high evaporation rates which might attain on the average up to 700 kg of steam per m^2 of heating surface per hour made it necessary to resort also on the water side to forced circulation by means of a circulating pump. As, however, the water paths are very short, the pump power remains low. The pump, therefore, could be made to deliver an additional head utilized for the purpose of steam separation from circulating water by means of centrifugal action. We were thus the first to introduce the centrifugal separator for steam generators and have brought the same to a high degree of perfection.

All this fundamental and original development work was carried out in the year 1931. There arose out of the reconstructed explosion boiler a steam generator operating with constant pressure, to which has since been given, in recognition of the principle employed, the name "Velox". The Velox principle combines the use of combustion under pressure, with very high flue gas velocities and with a gas turbine, whereby the pressure in the combustion chamber, the high gas velocities and the pressure drop of the gases for production of work are created by means of a compressor driven by the gas turbine, the latter being actuated by the products of combustion. The great significance of the Velox idea lies in the fact that by means of the increase in pressure and velocities the components are greatly reduced in size, and the processes are accelerated without impairing the efficiency. On the contrary, the latter is generally considerably improved. This fact is made possible by the gas turbine whose energy requirements first abstracted from the process are entirely restituted to it by means of the compressor.

Already the first Velox trials were crowned with success. Indeed the very first construction with 1 m^3 of furnace volume and a charging pressure of $2{\cdot}4$ kg/cm^2 abs achieved heat liberation rates of 7 million kcal/h and m^3, evaporation rates of over 500 kg/m^2 of steam per hour. The quick starting time of 6 minutes from cold to full pressure (28 kg/cm^2g) and full output (11 t/h) as well as the rapid regulation, provoked general astonishment (15).

For the further development and attainment of high degree of perfection of the Velox steam generator another problem had to be solved, which in the case of the first Velox boiler had been temporarily deferred, namely the improvement of the charging set. In the first trial boiler employed a charging set of the type used for Diesel engine supercharging was employed. The gas turbine consisted of a single impulse wheel mounted overhung on the shaft and the charger of the two-stage radial blower. In order to keep down the dimensions of the charging set, notwithstanding the large volumes to be delivered, and in order to be able to drive it without the addition of external power, that is to say only with the energy obtainable from the combustion gases, it was necessary to improve substantially the design and the efficiency of the gas turbine and compressor.

The axial compressor.

The solution of this problem was provided by the appearance of the axial compressor. As far back as 1927, we had taken up the manufacture of this type of compressor and at that time had already carried out exhaustive tests on a four-stage blower. The basis for the calculation and for the design of the blade shapes was given by the results of recent research in aerodynamics. This work was provided by tests on wind mills built by our French associates in Paris. It was from there that the incentive came to apply the aero-foil theory of aviation to the design of turbo-machinery. Numerous designs of single and multi-stage axial blowers for supplying the cooling air of large generators followed, and finally came the first large axial compressor for the supersonic wind tunnel at the Aero Dynamics Institute of the Federal Polytechnic School in Zürich (23).

It was the Velox boiler, however, which became the main application field of the axial compressor. Already the first unit to be sold, comprising a boiler of an output of 12 tons of steam per hour for the Mondeville Steel Works in France in 1932, was provided with axial compressors for the combustion air and the blast furnace gas. Since then some 80 axial compressors have been built for Velox boilers alone, among which are units of up to 1500 m^3/min suction volume, 3·5 kg/cm^2abs pressure, and requiring nearly 4000 kW to drive them. As will be seen later, the experience gained with the Velox and its combustion under pressure led to other important applications of the axial compressor.

The Velox and applications of the Velox principle.

The introduction of the Velox boiler was very rapid. This was all the more notable as it made its appearance during the worst industrial crisis and during which but few other installations were effected. Moreover, it was a new engineering development about which no experience was available and about which it was difficult to form an opinion. Already at the conclusion of the first year of its appearance, 1933, seven boilers were on order for a total output of 160 t/h, this figure rising at the end of 1935 to 31 units with a total output of 820 t/h, and at the end of 1940, excluding marine boilers, to 75 units at a total hourly output of 2500 tons (8, 16—18, 21).

The main application of the Velox was as a peak-load boiler in electric generating stations and for stand-by plants, where its quick starting characteristics are particularly valuable (22, 24, 27). Many plants were supplied, however, also for continuous service, both for electric generating and for industrial plants, where oil, blast furnace gas, or natural gas constitute the normal fuel. The application of Velox boilers to ships, for reasons in no way connected with the technical features of the Velox, has been much slower. This is all the more incomprehensible in that the Velox incorporates all those characteristics which are desirable of the ideal marine boiler.

The largest boilers built up to the present for both oil and blast furnace gas, deliver 100 tons of steam per hour. The dimensions of a 100-ton blast furnace gas boiler correspond to those of a 150-ton oil-fired boiler. Notwithstanding these large outputs, it is possible to load the entire combustion chamber with built-in evaporating heating surfaces on a special railway truck.

The use of the Velox principle imposes no limitations in regard to the steam pressure and the superheat temperature other than those applying to the design of any other modern boiler. The Velox has also been applied to locomotives, although in this case matters have not gone further than a single very satisfactory trial outfit (33).

It was already hinted above that not only the Velox, but also other applications of the axial compressor and of the gas turbine have been developed by us. We had in mind an important field in the chemical and metallurgical industry. As far back as 1922/23 we had occasion to build gas turbines in which power was developed by hot exhaust gases constituted by the products of chemical processes. It is, however, not of these plants which it is intended to speak, but of others, in which, for the purpose of increasing the output as well as reducing the size of the apparatus employed, the chemical process is conducted under pressure and the hot gases exhausted employed for the purpose of actuating the gas turbine, the latter driving the compressor, which produces the required increase in pressure for the process. It is another aspect of the Velox principle. In actual fact, the Velox boiler provided the incentive for the use of Velox charging sets in the chemical industry. The engineers of the Sun Oil Co. of Philadelphia were the first to recognize this and to employ our Velox charging sets for the regeneration of the catalyser in oil refining according to the Houdry process (25, 31). The first set to be delivered to the Sun Oil Co. consisted of a compressor unit for a suction volume of 1160 m^3/h and a delivery pressure of 4·2 kg/cm^2 abs. The power taken for driving the compressor was 4400 kW. When supplied with exhaust gases from the regenerator, the turbine delivered 5300 kW. Thus there was an excess of power of 900 kW with which electrical energy was developed. The starting motor served here as generator and was dimensioned accordingly.

In the charging process an excess of power is always to be reckoned with, when with high efficiencies of the compressor and gas turbine, the temperature of the exhaust gases attains at least 500° C and the pressure drop due to the flow through the apparatus is small, and moreover, the weight of the gases returning to the gas turbine is at least equal to that of the air supplied by the compressor. If the apparatus is replaced by a combustion chamber which is supplied with part of the air from the compressor, whilst the remaining greater part of the compressed air is mixed with the products of combustion in order to cool them, the charging set becomes a gas turbine plant for the production of energy only.

The above mentioned first Sun Oil plant may, therefore, be looked upon as the forerunner of the pure gas turbine. About 20 similar plants followed that delivered to the Sun Oil Co., and amongst them came the gas turbine for pure energy production. Thus an object which had been striven at during several decades, namely the creation of a serviceable gas turbine with the constant-pressure combustion, had at last been achieved.

The pure gas turbine with constant-pressure combustion.

The first pure gas turbine was installed at Neuchâtel in Switzerland. Prior to its installation on site, it constituted one of the show-pieces of the Swiss National Exhibition in Zürich in 1939 (9). The set is intended for stand-by purposes and delivers 4000 kW. The output of the turbine proper is 16,000 kW. The power consumption by the compressor is 12,000 kW. The compressor pressure is $4 \cdot 2$ kg/cm^2 abs, the temperature of the hot gases attaining at full load 550^0 C. The design details of the compressor and turbine are the same as those of the Velox charging sets. The only novel part was the combustion chamber. For this also constructional details could be employed which had been worked out for similar conditions, for we had already constructed and had had in service for a considerable time combustion chambers to take the place of heat exchangers and air preheaters in producing hot gases under pressure for the purpose of testing exhaust turbo-chargers.

Moreover, it was possible to apply to the combustion chamber the general construction principles worked out for the Velox boiler.

An important component of the gas turbine is the preheater in which the combustion air is heated by the exhaust gases. The recuperation of the heat of the exhaust gases can affect considerably the efficiency of the gas turbine plant. This means, however, usually very large heating surfaces, and we have carried out extensive investigations in order to determine the most favourable conditions in relation to heating surface, output and pressure drop due to flow.

In plants where simplicity and cheapness are the deciding factors and where fuel costs are not of considerable importance, the preheater may be dispensed with. With a gas temperature of 550^0 C it is possible to obtain with a unit consisting only of the gas turbine and the direct-coupled compressor an efficiency of about 16—17 %. This value is not high in comparison with those obtainable with a modern steam turbine plant, but it is sufficient for many applications such as stand-by and peak-load duty or similar plants required mainly for only short periods of service, or again in cases where cheap fuel, such as blast furnace gas, is available.

Velox and gas turbine for supplying blast requirements in steel plants.

For this reason we have given special attention to the generation of the power, compressed air and blast requirements and blast heating for steel works. Already in 1933 there was supplied the first Velox boiler for a blast furnace plant. Since then, large blast-furnace-fired Velox boilers have come into service abroad and further sets are under construction.

A very interesting application was found for the gas turbine in blast production by causing the compressed air and the gas turbine air to be delivered by the same compressor. The advantage of this combination lies not only in the cheapening and the simplification of the plant, but also in the increase of economy. The compressed air or the blast air quantity is always considerably less than the combustion and cooling air required for the gas turbine. Thus variations in the quantity and pressure of the blast furnace operation affect the total output of the compressor but little, so that instead of the centrifugal or radial type of compressor alone used hitherto for blast production, it is possible to employ the axial type compressor, unquestionably better from the efficiency point of view, notwithstanding its less favourable characteristics in regard to surging limit. By producing the gas turbine air and the compressed air in one and the same machine, it is possible to avoid also the losses associated with the transmission of energy, where special blast blowers are driven for example by means of steam turbines (19).

At the present time a number of gas turbine plants for blast production are under construction. The useful air quantity of each of these plants amounts to 100,000 Nm^3/h. The blast pressure is about 2 kg/cm^2 abs, and the pressure of the gas turbine air amounts to about 3 kg/cm^2 abs. The total output of the gas turbine which drives through a gear also the gas compressor for the blast furnace gas, is 10,000 kW, whilst for the compression of the blast air alone about 2500 kW are required. These gas turbines are equipped with recuperators, each of which is provided with a surface of about 4000 m^2. We expect to obtain with such a plant a gas consumption lower than that taken for the same blast conditions by gas engines driving reciprocating compressors.

Not only the Velox and the gas turbine, but also supercharging appears to us to be of considerable importance for the future developments of steel plants. Already in 1935 we attempted by suggesting supercharging, to introduce metal blast heaters to take the place of Cowpers. It was proposed to heat the blast in two stages, the first stage being preceded by a Velox boiler in order to reduce the temperature of the combustion gases to a value admissible for the metal heating surfaces. Between the first and the second blast heating stages it was proposed to introduce the gas turbine for driving the compressors producing the compressed air and blast furnace gas for the combustion. The steam produced by the Velox was intended for driving turbines coupled to blast blowers.

Another proposal referred to the supercharging of blast heaters, where instead of the Velox for lowering the exhaust gas temperature without loss, part of the gases were abstracted from the power gas stream before entering the gas turbine and returned again to the upper stage of the blast heater.

Supercharging and lossless letting down of the temperature are, however, best achieved when the blast is produced by a gas turbine, and the gas turbine is driven by the hot gases of the blast heater. Only a single combustion chamber is then necessary. The adjustment of the combustion temperature to the hot gas temperature required for the blast heater, as well as the adjustment of the hot gas temperature to the generally lower value admissible for the gas turbine is attained by mixing a suitable quantity of cooling air taken from the common compressor.

Such a blast installation promises high economy. The material required and the space requirements are small, and a further special advantage is that, due to its small size, it may be placed in the immediate neighbourhood of the blast furnace, thereby avoiding any appreciable pressure drop and radiation losses and facilitating operation.

The gas turbine locomotive.

As the last milestone in the development of the gas turbine and of supercharging there should be mentioned the gas turbine of a locomotive built in 1940 and which, after lengthy trials in the test department is at the present time (June 1940) being erected on the locomotive frame (35). The output is 2200 H.P., this output being transmitted electrically to the driving wheels. The fuel is gas oil, the temperature of the gases before the turbine is 570^0 C, and the fuel consumption is about 400 g/H.P.h. Due to its advantages, one of the main ones being the independence of any water supply, it is possible that there will be a big future for this type of locomotive, particularly in countries where water is scarce and oil is available.

Conclusion.

Finally, mention should be made of the tests being carried out on grinding and combustion of pulverized fuel and of dust elimination under pressure. Our tests in this direction were started already in November 1933. Up to the present success has been obtained with furnace heat liberation rates of the same order as those obtained with oil, as well as in the creation of a suitable burner. The complete solution involves, however, still clearing up the problem of the slag to be removed from the combustion chamber and of an adequate elimination of the dust from the gases going to the gas turbine. Because pressure-firing requires solutions of the regulation entirely different from those applied to firing systems operating at atmospheric pressure, new problems arose in regard to the regulation of the pulverized fuel and of the grinder.

Undoubtedly, the immediate employment of coal as fuel for the Velox boiler and the gas turbine would open out great possibilities. Our efforts to bring to a successful conclusion our labours on the subject of applying pressure-combustion to pulverized coal, may therefore be well understood. A further task of perfecting both Velox and gas turbine also lies before us. In the case of the Velox, development will be in the direction of a cheapening of the design and an improvement of the adaptation of the component parts to available space conditions, particularly in regard to application in ships.

The gas turbine is only on the threshold of its practical application and has large development possibilities in front of it. We are especially trying to increase its efficiency, without, however, detracting from the ideal simplicity of the plant. In addition, there are various already carefully studied and in part already developed arrangements and operating processes to be brought into such shape so as to be practically applicable, even though this may entail a departure from the simple design, if an improvement of the efficiency and an increase of output of the gas turbine unit is to be attained thereby, because an increase of efficiency and of output will greatly widen its field of application. Here again we have in mind its application to ships.

As regards the development of the charging of combustion engines. We expect to continue on the lines followed during the last few years.

The supercharging of appliances of the chemical and metallurgical industries will we hope receive the interest and the understanding from those circles without whose collaboration our best suggestions must remain unfruitful.

Throughout this review of 35 years of engineering development, the gas turbine in one form or other is continually cropping up. It is seen that not only have we examined every possibility of realizing this machine, but that we actually tried many of these possibilities. Thus, where we are concerned, the gas turbine rests on a sure foundation and does not represent a subject recently taken up in order not to miss a new development.

(MS 761) *Dr. W. G. Noack. (Hv.)*

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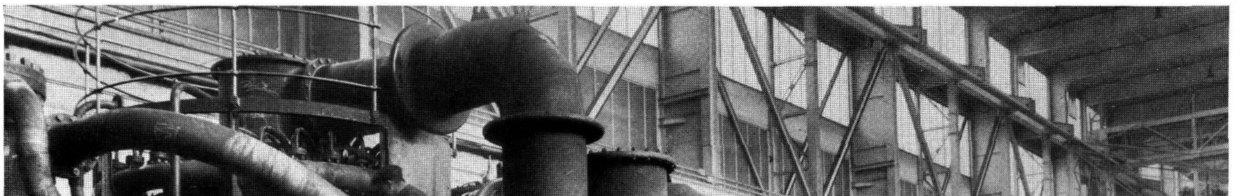


Two Velox boilers, evaporation 50 and 18 t/h, respectively, shortly before completion of assembly in our workshops. In background upper water chamber for a further 50 t/h Velox boiler.

Velox boilers are subjected to exhaustive works tests fully assembled before dispatch.

THE INFLUENCE OF MULTI-STAGE COOLING ON THE EFFICIENCY OF THE NEW BROWN BOVERI "ISOTHERM" TURBO-COMPRESSOR.

Decimal index 621.515.5—71

The compressors described in this article are of the centrifugal type. However, in conjunction with the gas turbine, another type of compressor has come into prominence to-day, namely the axial compressor. Despite this, the radial or centrifugal compressor has maintained its old pre-eminent position for all other applications.

The development of the cooling equipment for the compressor went hand in hand with that of the latter. Along with improvements applied to the air-flow conditions in the compressor, amelioration of the cooling is the most effective means of raising the efficiency of the compression process. For this reason, we have studied the cooling problem continually ever since we began building compressors in 1906. Starting from water-jacket cooling, we then find a mixed jacket and intermediate cooling system and then purely intermediate cooling first with three coolers, model 1915, and then with the multi-cooler arrangement, model 1935. Some of the illustrations allow of following this development. The mathematical proof is produced, with the help of an example, that the present-day multi-cooler design (with 7 coolers for example) as compared to the earlier one with 3 coolers allows of gaining 4 % in efficiency and that another 4 % are saved by improvements to the flow conditions resulting from the suitable location of the coolers. 8 % increase in efficiency, as compared to the 1915/34 design, allows of paying off the extra outlay for bigger cooling surfaces, within less than a year.

(a) *General considerations on the cooling of turbo-compressors.*

The fact that when compressing in a ratio of 1 to 8, for example, the adiabatic or uncooled compression requires 36 % more compression work than does the isothermal one, suffices to show up the importance to be attached to cooling. The turbo-compressor design allows of incorporating cooling devices because the big number of impellers necessary owing to the peripheral speed make it possible either to design the air tracks between the stages in the form of water jacket cooling or else to insert intermediate coolers on the said air tracks. These two types of cooling can also be combined.

At the beginning of compressor building, the material of which the impellers were made was not as good as it is now and it was necessary to have up to 25 impellers divided up between 2—3 housings. At that period, solely water-jacket cooling was applied. As an example of this class of machine the three-cylinder compressor built by us in 1906 and which was the first turbo-compressor in the world has a certain historical interest (Fig. 1). It was built to the data supplied by Prof. Rateau of Paris for the first gas turbine constructed by Lemale & Armangaud, for 60 m^3/min intake volume and only 3·5 kg/cm^2 delivery pressure; although it had 25 impellers. This pioneering effort was followed by a whole series of combined water-jacked cooled and intermediate cooled compressors built up till about the end of 1915. From that time, and thanks to better material being available, it was found possible to work to a pressure ratio of 1 to 8 with only 11 impeller wheels which could all be lodged in one housing. This, of course, reduced the surface available for water-jacket cooling which became insufficient so that it had to be replaced by external cooling surfaces. Thus, we find a design with 3 intermediate cooler pairs, the coolers being placed in V-formation on each side of the compressor housing.

The entropy diagram (Fig. 2) proves clearly that already with intermediate cooling twice repeated the simultaneous utilization of water jacket cooling offers no more interest. The figure shows the heat carried off by water jacket in the case of jacket cooling alone (Q_0), for jacket cooling and 2 intermediate coolers (Q_2) and for jacket cooling and 5 intermediate coolers (Q_5). These are the shaded surfaces below the compression characteristics a_0, a_2 and a_5. The effectiveness of jacket cooling drops very sharply as the number of intermediate coolers increases, this because the intermediate coolers deprive them of a considerable part of the temperature drop.

It can be asserted that the 1915 model with 3 intermediate coolers and without jacket cooling (Fig. 3) was very advanced from the point of view of the technology of cooling. Despite this progress, our firm made another big stride by increasing considerably the number of coolers, in 1935. By increasing the number of coolers to 7 and thus attaining repeated recooling, the question was, however, not exhausted. The aim is to have a big number of coolers so as to make it possible to effect cooling after every impeller and so to design these coolers that the inflow and outflow of air is as favourable as regards flow conditions as is possible. Further efforts have been made to put in considerably more cooling surfaces than had been thought possible or was common practice before, so that speeds could be made lower

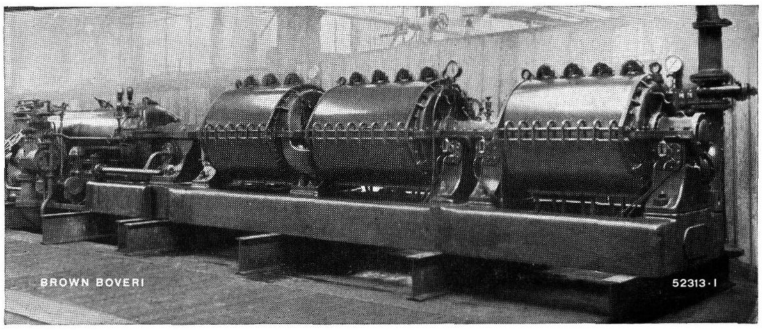


Fig. 1. — Three-cylinder compressor with water-jacket cooling and 25 impellers built in 1906 (without intercooling).
This was the first multi-stage turbo-compressor in the world.

and flow losses reduced. It was no light task to fulfil all these requirements simultaneously.

The turbo-compressor built according to these principles has been given the name "Isotherm" because it approaches a compression process at constant temperature.

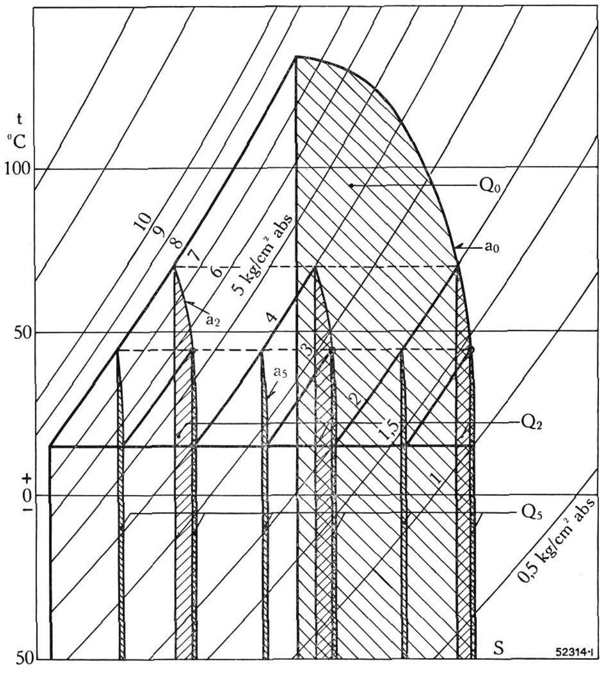


Fig. 2. — Heat carried off by water-jacket cooling.
Abscissae:— Entropy S. Ordinates:— Temperature T.
Q_0. Without intermediate cooling.
Q_2. With two intermediate coolers.
Q_5. With five intermediate coolers.
a_0, a_2, a_5. Lines of compression of the jacket-cooled stages.
As the number of intermediate coolers increases, the amount of heat dissipated by the jacket cooling gets less and less (shaded surfaces); thus, it is no longer worth putting it in.

(b) *Thermal comparison between a 3-stage cooling equipment and a 7-stage one, for example, assuming total recooling to initial temperature.*

The work of compression performed per kg of air and for z cooling stages, that is for (z + 1) impeller stages, is

$$L = \frac{R T_0}{\frac{k-1}{k}} \left[\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right] (z+1) \frac{1}{\eta_{ad}}$$

here R is the gas constant, T_0 the temperature at the suction branch, k the adiabatic exponent of compression, p_2/p_1 the pressure ratio per stage and η_{ad} the adiabatic efficiency of the stages. With the temporary assumption that the air is entirely recooled to 15^0 C, and allowing an efficiency of the stages of 75 % and a pressure ratio of 8 to 1 we reach the following comparative values:—

TABLE I.

Number of intermediate coolers	z	3	7
Number of compression stages	z + 1	4	8
Temperature t_0 and T_0 respectively	°C	15/288	15/288
Pressure ratio per compression stage	p_2/p_1	1·6818	1·2967
Work of compression L . . .	mkg/kg	25,300	24,300

The increased cooler number from 3 to 7 produces a thermal gain of

$$\frac{25{,}300 - 24{,}300}{25{,}300} \cdot 100 = 4\,\%.$$

If recooling is not complete, a correction must be introduced because the subsequent z compression stages following the first one demand more compression work as a result of the original temperature T_0 at which the air was drawn in, being now higher by Δ t. The compression work is increased in the ratio

$$\frac{T_0 + \Delta t}{T_0}$$

TABLE II.

Number of intermediate coolers	z	3	7
Temperature of cooling water at inlet . . . . . . t_{w_1}	°C	15	15
Recooling of the air down to t_3	°C	26	28
Difference of temperature $t_3 - t_{w_1}$	°C	11	13
Corresponding increase of work of compression $\Delta L' = \frac{z}{z+1} \cdot \frac{t_3 - t_{w_1}}{288} \cdot 100$	%	2·86	3·96
Corrected work of compression $L' = L + \Delta L'$	mkg/kg	26,025	25,265

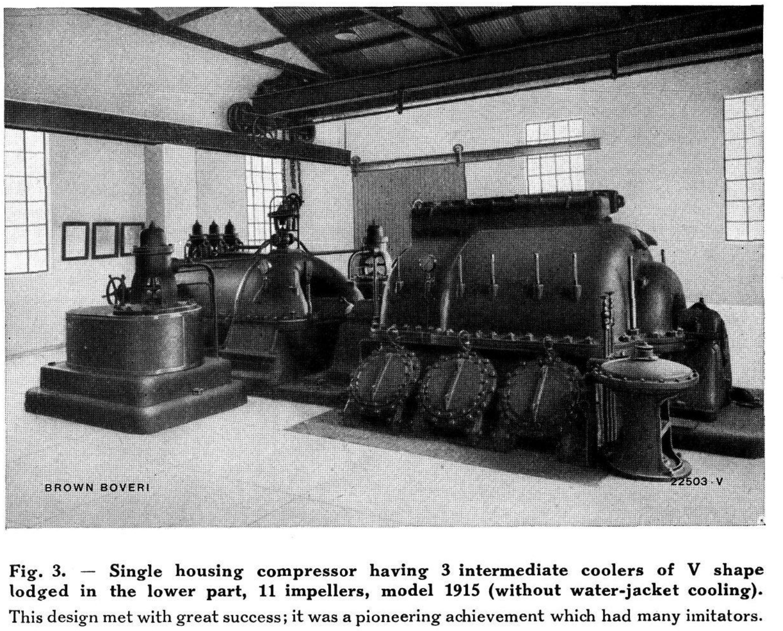


Fig. 3. — Single housing compressor having 3 intermediate coolers of V shape lodged in the lower part, 11 impellers, model 1915 (without water-jacket cooling).
This design met with great success; it was a pioneering achievement which had many imitators.

(c) *Influence of incomplete recooling.*

With the recooled temperature t_3 which is 2^0 higher in the compressor with 7 intermediate coolers, $1 \cdot 1$ % of the 4 % given by Table I are lost, according to Table II, so that there are $2 \cdot 9$ % left in favour of 7 stage cooling. Now, in itself, this is a considerable improvement but does not take the pressure drop in the coolers into account. The latter is given in the next paragraph. The compression process with both the numbers of cooler considered is shown in the entropy diagram of Figs. 4 and 5.

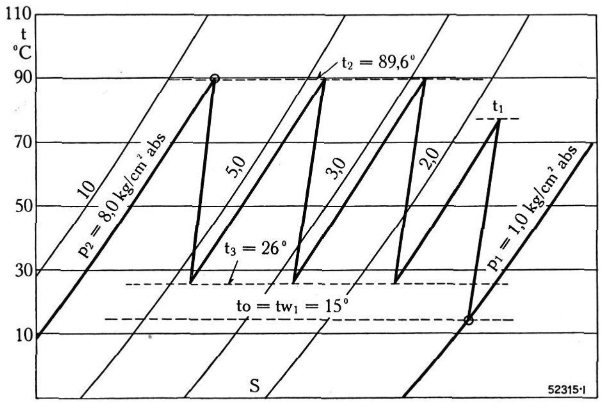


Fig. 4. — Entropy diagram of intermediate cooling in three stages.
Abscissae: Entropy in S. Ordinates: Temperature in t.

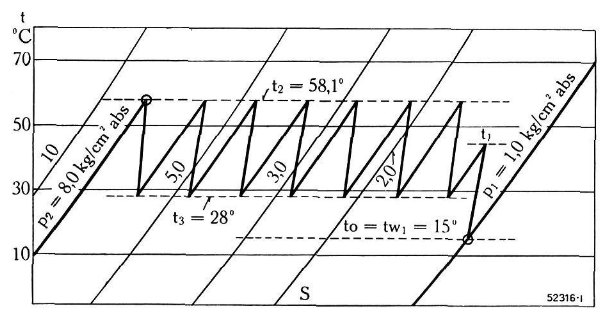


Fig. 5. — Entropy diagram of intermediate cooling in seven stages.
Abscissae: Entropy in S. Ordinates: Temperature in t.
The seven-stage intermediate cooling reduces considerably the work of compression as compared to three-stage cooling (see Table II).

(d) *Influence of the pressure drop in the cooler.*

As already mentioned, the speeds were reduced on account of the influence of the pressure drop in the cooler; with 7-stage cooling the speeds are only about half their value with 3-stage cooling. The heat transmission coefficient is reduced to about $^2/_3$ of its value by this drop in speed but the cooling surface is increased in still greater proportion. As the temperature peaks are not so high in the case of big numbers of coolers (Figs. 4 and 5) it is possible to make the coolers shallower. The pressure drop is a function of the square of the speeds (w_o^2) and of the depth of the coolers (T).

TABLE III.

Number of coolers	z	3	7
Work of compression L' according to Table II	mkg/kg	26,025	25,265
Corresponding heat value A L'	kcal/kg	61	59·2
Heat value per stage $\frac{A \cdot L'}{z+1}$.	kcal/kg	15·25	7·40
Temperature increase per stage $\frac{A \cdot L'}{(z+1) \cdot c_p}$	°C	63·6	30·1
Depth of cooler T	value ratio in %	100	47·6
Velocity of air w	value ratio in %	100	50
Therefore pressure drop $\Delta p \simeq w^2 \cdot T$	value ratio in %	100	12
Real pressure drop	mm water gauge	600	72
Average volume v_m during compression	m³/kg	0·215	0·205
Therefore loss work $\Delta L'' = \frac{v_m \cdot \Delta p}{\eta_{ad}} \cdot z$	mkg/kg	520	140
or $\Delta L''$ in % of L' . . .	%	**2·0**	**0·56**
Resulting work of compression $L'' = L' + \Delta L''$	mkg/kg	26,545	25,405

The difference in the loss due to pressure drop in the 7 coolers as compared to 3 coolers is $1 \cdot 44$ % despite the greater number of coolers because of the speed of the air being reduced to a half and because the depth of the coolers is about half, so that we get a total difference of compression work of $4 \cdot 34$ %.

TABLE IV.

Number of coolers	z	3	7
Heat value of work of compression AL'	kcal/kg	61	59·2
Temperature of air before cooler t_2	°C	89·6	58·1
Temperature of air after cooler t_3	°C	26	28
Temperature of water before cooler t_{w_1}	°C	15	15
Temperature of water after cooler t_{w_2}	°C	25	25
Temperature difference $\Delta t = \frac{(t_2 - t_{w_2}) - (t_3 - t_{w_1})}{\log_e \frac{t_2 - t_{w_2}}{t_3 - t_{w_1}}}$	°C	30·4	21·5
Recooling temperature drop $t_2 - t_3$	°C	63·6	30·1
Per stage heat carried off $Q = c_p \cdot (t_2 - t_3)$	kcal/kg	15·25	7·23
Therefore necessary for each step $k \cdot F = \frac{Q \cdot 3600}{\Delta t}$ . . .	$\frac{\text{kcal/h}}{{}^0\text{C}}$	1805	1210
For all stages $z \cdot k \cdot F$ . . .	$\frac{\text{kcal/h}}{{}^0\text{C}}$	5415	8470
Are utilized k kcal/m²/°C/h .		150	100
Therefore per kg/s required cooling surface	m²	**36·1**	**84·7**

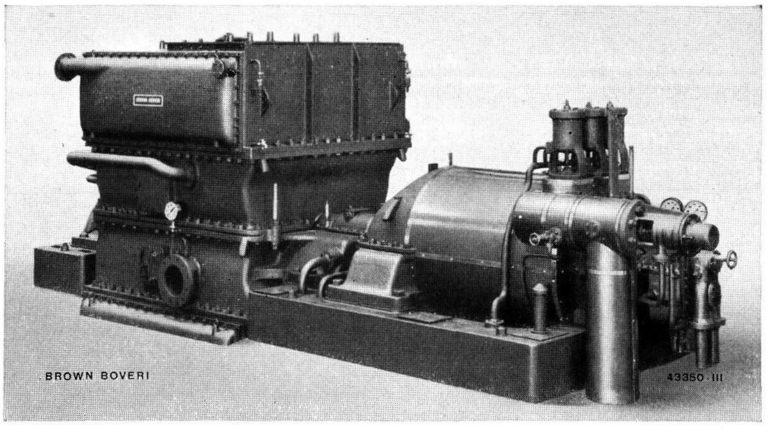


Fig. 6. — Single housing compressor having 7 intermediate coolers built on to the lower and upper housing and 9 impellers, model 1935 (without water-jacket cooling). These latest Isotherm turbo-compressors are characterized by an efficiency never attained so far.

the cooler are not of importance thanks to the low velocity of the air.

The gains attained thanks to improved flow conditions amount to about 3·5 at 4%. With the improvement of efficiency calculated under (d) and amounting to 4·34%, we now get an overall improvement of 8 %. In the compressors built up till now, model 1935 of medium size, isothermal efficiencies of nearly 70 % were repeatedly recorded with a cooling-water temperature equal to that of the air at the suction branch of the compressor. The bigger compressors can confidently be expected to better this figure of 70 %.

Of course, it would be possible to reduce the pressure drop in 3-cooler compressors to the value assumed here for the 7-cooler compressor by using lower speeds. The bigger cooling surfaces which are then necessary would, however, make direct building-on of the coolers according to Fig. 3 an impossibility.

(e) *Necessary cooling surface.*

The lower speed in the coolers demands more cooling surface to carry away the given quantity of heat. Table IV gives the calculations necessary to determine this surface.

The improvement found under (d) amounting to 4·34% thus calls for a cooling surface 2·35 times bigger.

(f) *Construction and design of the coolers.*

Fig. 6 shows the external appearance of the compressor.[1] The cooling tubes are lodged in two housings one being flanged on to the upper part of the compressor housing and the other to the lower one. These closely finned tubes can be removed separately. They lie horizontally in the cooler housing. The latter are built up of welded sheet metal. The flow conditions between impeller and cooler have been the subject of special attention and numerous models were made to investigate the said conditions. By means of guide blades which, as it were, are prolongation of the diffusor blades, the transition from speed into pressure has been improved considerably. The diametrical position of the two cooler halves of each stage and the width of the coolers allow of avoiding filling troubles on the impellers to a great extent. Further, the losses due to reversion of sense of flow when the air leaves

[1] See The Brown Boveri Review, 1941, No. 4, page 108.

(g) *Economic advantages of increased cooling surfaces.*

The cooling surface increase found under (e) of about 50 m^2 per kg of air drawn in per second costs Fr. 2500.—, at a price of Fr. 50.— per m^2 of cooling surface including the cooler housing. As compared thereto, an 8% improvement in efficiency, for a total compression work of 250 kW for 1 kg of air drawn in and compressed to 8 kg/cm^2 gauge, amounts to an annual saving of $250 \cdot 0{\cdot}08 \cdot 8000 = 160{,}000$ kWh or, at 2 centimes per kWh a saving of Fr. 3200.—. Thus the extra financial outlay is covered in less than a year. This shows unequivocally the great importance of this improved cooling method.

(h) *Conclusions.*

Some firms still adhere to water jacket cooling even to-day. The undisputable advantages of water jacket cooling are advantageous flow conditions and there being no sharp deviations in the direction of flow which is characteristic of this design when the housings are of big diameter. These problems had to be studied and solved when intermediate cooling was introduced. One important advantage of intermediate cooling is the much bigger cooling surface which can be accommodated so that the specific cooling work (k·F), for example, is as much as 3 times greater than with water-jacket cooling in the case of the 7-stage cooling described even with the biggest possible housing diameters and largest number of guide vanes.

The fact that, during the last 5 years, nearly 100 machines of the 1935 model with intakes of each 8000 to 80,000 m^3/h were ordered for new plants or as repeat orders is the best proof of the qualities of this new "Isotherm" turbo-compressor.

(MS 762) *Ad. Baumann. (Mo.)*

THE INFLUENCE OF COMPRESSIBILITY OF THE FLUID ON THE PROPERTIES OF A CENTRIFUGAL BLOWER.

Decimal index 621.515.5.001

The compressibility of the fluid has a considerable influence on the properties of a centrifugal blower, especially at the higher speeds now in use.

This influence depends firstly on the ratio of peripheral speed to the velocity of sound in the fluid, i. e. on the number of Mach $M = u_2/a_0$. Secondly also the adiabatic coefficient $k = c_p/c_v$, i. e. the ratio between the specific heats at constant pressure and constant volume, has a certain influence. The bigger M and the smaller k the greater is the influence of compressibility. As heavy gases and vapours (modern refrigerants, petrol, vapour, etc.) have low sound velocities and low adiabatic coefficients, compressibility has most influence with such fluids.

The following points have been investigated:—

(a) The influence of compressibility on the power consumption of a blower wheel.
(b) The limit volume.
(c) The influence of compressibility on diffusor dimensions.

IN modern high-speed blowers the air is already considerably compressed in the wheel itself. The characteristic of the wheel is thereby changed to an appreciable extent and blowers differ in this point from low-speed ventilators in which, as in pumps, the fluid has practically always the same volume.

According to the laws of similarity, the compressibility is expressed by the number of Mach, i. e. by the ratio of any velocity to the velocity of sound in the fluid. With blowers, it is convenient to compare the *peripheral velocity* u_2 to the *velocity of sound* a_o *in the fluid at rest on the suction side* of the blower. We consequently define:

$$M_0 = \frac{u_2}{a_o} \tag{1}$$

In principle, the influence of the number of Mach can only be found by experiment. With certain assumptions, however, some of the consequences of compressibility may also be calculated.

(a) *Influence of compressibility on the power consumption of a blower wheel.*

The theoretical manometric head of the blower is $H_o = L/G$ (L = power; G = quantity), from which a part $H_2 = \varrho_o \cdot H_o$ is transformed into pressure in the wheel itself, with the efficiency $\eta_R \cdot \varrho_o$ is the theoretical degree of reaction and may be calculated from observations[1]. From this, the temperature T_2 at the outer circumference of the wheel is found:

$$\frac{T_2}{T_o} = 1 + \frac{k-1}{k} \cdot \frac{\varrho_o H_o}{RT_o} \tag{2}$$

[1] See "The separation of impeller and diffusor losses in radial blowers" on page 203 of this number.

and with polytropic compression the volume in the wheel diminishes in the proportion

$$\frac{v_o}{v_s} = \left[1 + \frac{k-1}{k} \cdot \frac{\varrho_o H_o}{RT_o}\right]^{\frac{1}{n-1}} \tag{3}$$

n is the polytropic coefficient of compression in the wheel, which can also be replaced by the adiabatic coefficient k and the wheel efficiency η_R, according to the equation $(k-1)/k = \eta_R \cdot (n-1)/n$. If, instead of the theoretical manometric head H_o, a

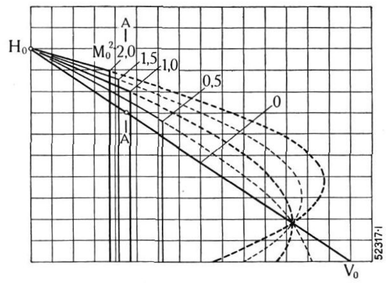


Fig. 1. — Theoretical manometric height of a wheel with different numbers of Mach.

H_o. Theoretical manometric height.
V_o. Suction volume.
A—A. Normal volume.

Based on the straight line for $M_o^2 = 0$, the lines for $M_o^2 = 0{\cdot}5;\ 1{\cdot}0;\ 1{\cdot}5$ and $2{\cdot}0$ were calculated.

At the limit volume, which may be taken from Fig. 2, these lines break down vertically. The real manometric height is found by multiplying the theoretical manometric height by the efficiency.

dimensionless power coefficient μ_o be introduced according to the definition $H_o = \mu_o \cdot u_2^2/g$ and if it be further considered that $u_2^2/kgRT_o = u_2^2/a_o^2 = M_o^2$, the following equation is found:

$$\frac{v_o}{v_s} = \left[1 + \frac{k-1}{k} \cdot \frac{\varrho_o H_o}{RT_o}\right]^{\frac{k}{k-1} \cdot \eta_R - 1} = \left[1 + (k-1)\, \varrho_o \mu_o M_o^2\right]^{\frac{k}{k-1} \cdot \eta_R - 1} \tag{4}$$

The equation shows that the compression ratio is in fact only a function of the number of Mach, respectively of the equivalent expression H_o/RT. As it was assumed that conditions at the circumference and consequently the power consumption, respectively the theoretical manometric head H_o, will not change as long as the volume V_s at the outer circumference of the wheel remains constant, the suction volume v_o

must be increased in the ratio v_o/v_s. If, therefore, the curve $H_o = f(V_o)$ is known for an incompressible fluid, the corresponding curve for the compressible fluid will be found by deplacing every point at the same value of H_o in the proportion v_o/v_s in the direction of the V_o axis.

Fig. 1 shows the influence of compressibility. The power consumption increases considerably with the number of Mach. At the limit volume the curves break down vertically, the dotted parts cannot be realized in practice.

(b) *Limit volume.*

With high-speed compressors for heavy gases it sometimes occurs that the pressure volume curve breaks down almost vertically, a little above the normal point, because the speed of sound is attained in the wheel entrance and, therefore, no bigger volume can enter the wheel. This limiting volume may be calculated in the following way:

Let w_1 be the relative entrance velocity and u_1 the corresponding peripheral speed. Then the increase of energy in the entrance of the wheel is:

$$H_1 = (u_1^2 - w_1^2)/2g \qquad (5)$$

As soon as $w_1 > u_1$, expansion takes place in the entrance, which increases the specific volume and diminishes the temperature and consequently the velocity of sound.

The following equations then hold:

$$\frac{T_1}{T_o} = 1 + \frac{k-1}{k} \cdot \frac{H_1}{RT_o} = 1 + (k-1)\frac{(u_1^2 - w_1^2)}{2a_o^2} \qquad (6)$$

and

$$\frac{v_o}{v_1} = \left[1 + (k-1)\frac{(u_1^2 - w_1^2)}{2a_o^2}\right]^{\frac{1}{n-1}} \qquad (7)$$

If the speed of sound is attained in the entrance, then $w_1^2 = a_1^2 = a_o^2 \cdot T_1/T_o$, and consequently:

$$\frac{T_1}{T_o} = \frac{2 + (k-1) \cdot u_1^2/a_o^2}{k+1} \qquad (8)$$

and

$$\frac{v_o}{v_1} = \left[\frac{2 + (k-1) \cdot u_1^2/a_o^2}{k+1}\right]^{\frac{1}{n-1}} \qquad (9)$$

The maximum quantity which may enter the wheel per unit section of free entrance, is a_1/v_1, whereas the corresponding quantity with incompressible fluid would be $u_1/v_o \cos \alpha_1$ with shockless entrance under the angle α_1.

The ratio of limit volume to the shockless normal volume with incompressible fluid consequently is:

$$\frac{V_{max}}{V_o^*} = \frac{v_o}{v_1} \cdot \frac{a_1}{u_1} \cdot \cos \alpha_1 \qquad (10)$$

or with $a_1^2 = a_o^2 \cdot T_1/T_o$ and $u_1 = u_2 \cdot r_1/r_2$

$$\frac{V_{max}}{V_o^*} = \frac{\cos \alpha_1}{M_o\, r_1/r_2}\left[\frac{2 + (k-1)(M_o \cdot r_1/r_2)^2}{k+1}\right]^{\frac{n}{n-1} - \frac{1}{2}} \qquad (11)$$

If losses must be considered, $(n-1)/n$ would have to be replaced by $\eta_1 (k-1)/k$, as the fluid expands in the entrance when operating with limit volume if u_1 is smaller than a_1. Calculation shows that the ratio V_{max}/V_o may become smaller than unity.

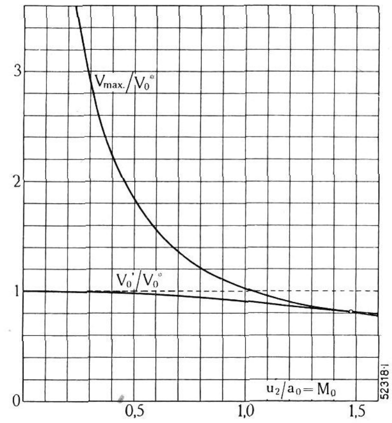


Fig. 2. — Variation of the limit volume and the volume entering without shock with the number of Mach.

V_{max}. Limit volume.
V_o'. Volume entering without shock with compressible fluid.
V_o^*. Volume entering without shock with incompressible fluid (i. e. "Normal volume").

The limit volume decreases considerably with increasing number of Mach and in the present example reaches the "normal volume" at $M_o \sim 1$. It may be increased by a suitable design of the wheel.

The reason is that the quantity which may enter the wheel without shock decreases with increasing number of Mach. Let V_o' be the entrance volume without shock, referred to suction conditions. Then

$$\frac{V_o'}{V_o^*} = \left[1 - {}^1/_2 \cdot (k-1) \cdot tg^2\, \alpha_1 \cdot (M_o \cdot r_1/r_2)^2\right]^{\frac{1}{n-1} - 1} \qquad (12)$$

This decrease in volume, entering without shock, is comparatively small and is only of importance in high-speed refrigerating compressors.

In Fig. 2 the ratio of the limit volume to the normal volume entering without shock and with incompressible fluid is shown in function of the number of Mach. When M_o approaches unity, the limit volume has already decreased to the normal volume, when M_o is bigger than unity, the normal volume cannot be attained at all.

The ratio of volumes that may enter the wheel without shock with compressible and incompressible fluid is also shown, for $M_o \sim 1$, the decrease is about 10 %.

(c) *Influence of compressibility on diffusor dimensions.*

At constant suction volume the diffusor must be closed considerably with increasing compression, i. e. with increasing number of Mach. The volume leaving the wheel diminishes considerably and also in the intermediate space between wheel and diffusor blading compression will steadily increase. For wheels with falling characteristic, the absolute outlet volume and consequently the diffusor must be still more closed.

If H_{2kin} is the kinetic energy at the wheel outlet, then in an unbladed diffusor with parallel walls, the fraction $H_{2kin}\left[1-(r_2/r_3)^2\right]$[1] will be converted into

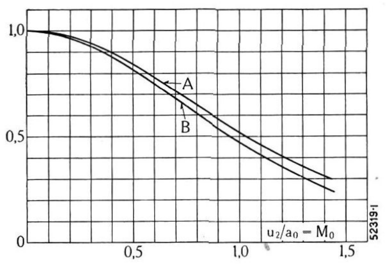


Fig. 3. — Variation of diffusor section with the number of Mach.
With increasing number of Mach the *volume entering* the diffusor diminishes considerably and with wheels having blades bent backwards, the angular momentum increases. For both reasons the diffusor sections must be diminished. The figure shows that this decrease may be considerable; in the present example it is about 50 % for $M_o \sim 1$.

A: For delivery of normal suction volume.
B: When volume enters wheel without shock.

pressure if r_2/r_3 represents the ratio of wheel radius to the entrance radius in the diffusor.

Analogously to equation (4), the compression ratio between wheel and diffusor then becomes:

$$\frac{v_2}{v_3}\left\{1+(k-1)\cdot\frac{1-(r_2/r_3)^2}{1+(k-1)\cdot\varrho_o\,\mu_o\,M_o^2}\cdot\frac{(1-\varrho_o)\cdot\mu_o\cdot M_o^2}{\eta_s}\right\}^{\frac{1}{k-1}\eta_s-1} \qquad (13)$$

with η_s = diffusor efficiency between wheel circumference and diffusor blading.

The total compression until entrance in the diffusor is then $v_o/v_3 = v_2/v_3 \cdot v_o/v_2$, and with constant tangential component of the absolute outlet velocity the diffusor must be closed in this ratio with increasing number of Mach.

Fig. 3 shows the result of calculation in one special case, the theoretical manometric height being taken from Fig. 1. The decrease in diffusor dimensions is considerable and approaches 50 % for $M \sim 1$. A still greater decrease is necessary if it is considered that according to Fig. 2 the volume which may enter the wheel without shock, also diminishes with increasing number of Mach.

In this connection the question may be raised, under which *conditions* the local velocity of sound is attained in the entrance of the diffusor blading. The velocity of sound in the entrance of the diffusor is

$$a_3^2 = a_o^2 + g\cdot(k-1)(H_o - H_{3kin}) \qquad (14)$$

and on the other hand the velocity of entrance is

$$c_3^2 = 2gH_{3kin} \qquad (15)$$

The velocity of sound is attained if $c_3^2 = a_3^2$, that is if

$$a_o^2 = (k+1)\,gH_{3kin} - (k-1)\,gH_o \qquad (16)$$

For H_{3kin} the following equation holds:

$$H_{3kin} = \left[1-\frac{1-(r_2/r_3)^2}{\eta_s}\right]\cdot(1-\varrho_o)\cdot\mu_o\cdot H_o \qquad (17)$$

and as with normal dimensions

$$1-\frac{1-(r_2/r_3)^2}{\eta_s} \sim {}^2/_3,$$

the condition that the local velocity of sound is attained in the diffusor entrance, is:

$$M_o^2 \sim \frac{1}{\mu_o\left[{}^2/_3\cdot(k+1)(1-\varrho_o)-(k-1)\right]} \qquad (18)$$

In an ordinary blower for air, the velocity of sound is never attained in the diffusor entrance. With wheels with radial blading the limit lies at $M_o \sim 1{\cdot}6$, that is in the neighbourhood of $u_2 = 500$ m/s. Refrigerating compressors for heavy refrigerants with $k \sim 1$, attain velocity of sound in the diffusor entrance at about the same value of $M_o \sim 1{\cdot}6$, i. e. at a peripheral velocity of about 200 — 250 m/s, if the much lower speed of sound in these refrigerants is considered.

(MS 769) *A. Meldahl. (Mo.)*

[1] In an unbladed diffusor, the full pressure is produced to begin with, the friction losses at first only appearing as a decrease in kinetic energy.

THE SEPARATION OF IMPELLER AND DIFFUSOR LOSSES IN RADIAL BLOWERS.

Decimal index 621. 515. 5-155-001

The efficiency of the impeller of a blower is defined here by making a comparison between a real blower and an ideal one. It is shown that this definition leads to practical results and that the efficiency so defined can be deduced from simple measurements.

Measurements carried out on an existing blower help to explain how a blower can be improved by carrying out separate measurements of its impeller and diffusor efficiencies.

IT is not easy to establish a true definition of what the impeller efficiency really is, in the case of a radial blower. The impeller certainly absorbs all the power supplied but only converts a part of it into pressure while the kinetic energy it delivers is converted into pressure in the diffusor.

By means of the following consideration, it is possible to reach a workable definition which has also the advantage that the efficiency thus defined is relatively easy to measure.

A blower generates pressure. Now assuming the same power input and the same delivery quantity, the better the blower the higher the pressure it will generate. Thus, it seems best to compare the real blower with an ideal one, without losses, delivering the same quantity and absorbing the same power. According to Euler's formula, the tangential component of the absolute delivery velocity at the periphery of the impeller then is the same in the ideal and in the real blower. If the ideal blower be so dimensioned that the radial component is the same as well, the kinetic energy in the gap will be identical for real and for ideal blower.

It is easy to calculate the pressure generated in the ideal blower. The total theoretical delivery head is $H_o = L/G$ (L = power input; G = weight delivered per unit of time); with inflow without angular momentum, the tangential component of the absolute delivery velocity at the impeller periphery, is $c_{2u} = gH_o/u_2$ (u_2 = peripheral speed); the radial component is $c_{2r} = G \cdot v_s / \pi D_2 b_2$ (v_s = specific volume in gap, D_2 and b_2 = diameter and width of impeller). Thus, the velocity head in the gap is $H_{2kin} = {}^1\!/_2\, g \cdot (c_{2u}{}^2 + c_{2r}{}^2)$ and the delivery head created in the ideal impeller is $H_2 = H_o - H_{2kin}$.

The real blower generates a delivery head H and, therefore, the efficiency of the blower is $\eta = H/H_o$. *This delivery head can be divided up between the impeller and the diffusor when the static gap pressure on the impeller periphery is measured.* Let the actual delivery head up to the gap be H_s. Then the diffusor efficiency is obviously

$$\eta_D = \frac{H - H_s}{H_{2kin}}$$

Analogously the impeller efficiency will be *defined* as

$$\eta_R = \frac{H_s}{H_2} = \frac{H_s}{H_o - H_{2\,kin}} \tag{1}$$

This definition leads immediately to

$$\eta = \eta_R \cdot \varrho_o + \eta_D\ (1 - \eta_o) \tag{2}$$

here $\varrho_o = H_2/H_o$ is the theoretical degree of reaction. This simple equation (2) for η shows that the definition according to (1) is a useful one.

With the help of the theoretical degree of reaction, the efficiences for impeller and diffusor can be expressed as

$$\eta_R = \frac{H_s}{\varrho_o\, H_o} \qquad \eta_D = \frac{(H - H_s)}{(1 - \varrho_o)\, H_o} \tag{3}$$

It is advantageous to introduce here dimensionless characteristic numbers, all delivery heads being referred to u_2^2/g. Then $H_o = \mu_o \cdot u_2^2/g$, $H = \mu \cdot u_2^2/2\,g$ and $H_s = \mu_s \cdot u_2^2/g$, which leads to

$$\eta_R = \frac{\mu_s}{\varrho_o\, \mu_o} \qquad \eta_D = \frac{(\mu - \mu_s)}{(1 - \varrho_o)\, \mu_o} \tag{4}$$

ϱ_o has still to be found. According to the Euler formula $c_{2u} = gH_o/u_2 = \nu_{2u}\, u_2$ for inflow without angular momentum. The Rateau volume figure is introduced for the volume according to the equation $V = \delta \cdot {}^1\!/_4 \cdot D_2^2 \cdot u_2$, therefore $c_{2r} = \delta_s \cdot \pi/4 \cdot D_2/b_2 \cdot u_2 = \nu_{2r} \cdot u_2$ (δ_s is referred to the volume in the gap) and this leads to

$$\mu_o\, \varrho_o = \mu_o - {}^1\!/_2\ (\nu_{2u}{}^2 + \nu_{2r}{}^2) \tag{5}$$

or

$$\varrho_o = 1 - \frac{(\nu_{2u}{}^2 + \nu_{2r}{}^2)}{2\,\mu_o} \tag{6}$$

If the inflow is without angular momentum, Euler's formula leads to $\nu_{2u} = \mu_o$ and then both expressions can be graphically expressed very simply in a system of coordinates ν_{2u}/ν_{2r} (Figs. 1 and 2). The lines $\mu_o\varrho_o$ = constant are concentric circles round the point ($\nu_{2u} = 1$; $\nu_{2r} = o$) and with a radius $r = \sqrt{1 - 2\mu_o\varrho_o}$. The curves ϱ_o = constant are also circles which all pass through the zero point ($\nu_{2u} = o$; $\nu_{2r} = o$) and the centres of which lie on the ν_{2u} axis; the radii are $r = 1 - \varrho_o$.

ν_{2r} is proportional to the delivery volume in the gap, therefore, the $\nu_{2u} - \nu_{2r}$ curve at inflow without angular momentum is nothing else than the dimension-

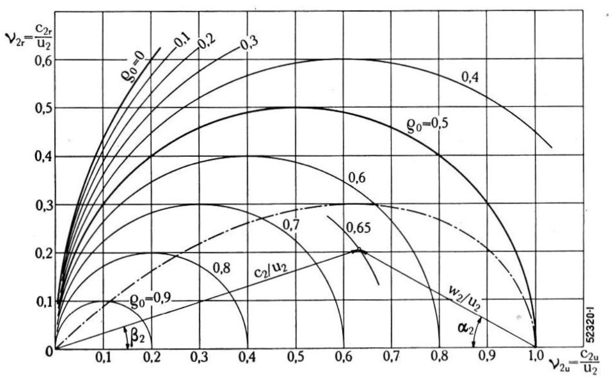


Fig. 1. — $\nu_{2u} - \nu_{2r}$. **Diagram with lines of equal degree of reaction in the case of inflow devoid of angular momentum.**

The dash-dot-line is that of maximum degree of reaction for a given discharge angle β_2.

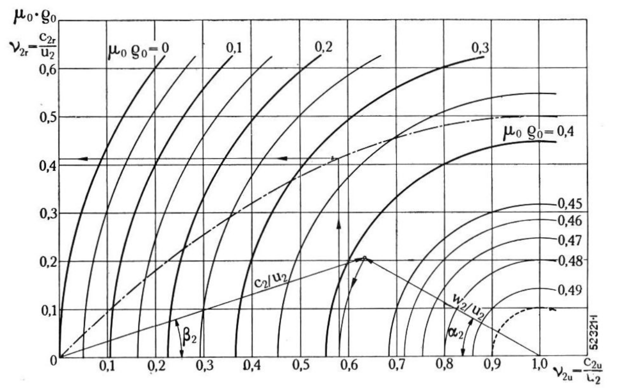


Fig. 2. — $\nu_{2u} - \nu_{2r}$. **Diagram with lines of equal theoretical pressure in the gap, with inflow devoid of angular momentum.**

The theoretical pressure in the gap $\mu_0 \varrho_0$ can also be read off the left-hand scale as shown by the arrows.

$\nu_{2u} = c_{2u}/u_2$. Tangential component of the absolute velocity of discharge.
$\nu_{2r} = c_{2r}/u_2$. Radial component of the absolute velocity of discharge.
α_2. Absolute angle of discharge.
β_2. Relative angle of discharge.

The diagram is a dimensionless representation of the discharge triangle.

less theoretical pressure-volume curve $\mu_o = f(\delta_s)$ somewhat deformed in the direction ν_{2r}. The curves $\mu_o \varrho_o$ = constant and ϱ_o = constant can be transferred directly to the pressure-volume diagram, the circles becoming ellipses. The $\nu_{2u} - \nu_{2r}$ diagram, it may be mentioned in passing, is nothing else than a dimensionless representation of the outflow triangle.

The calculation of the efficiency of the impeller from the pressures measured, power inputs and delivery quantities is then carried out as follows: In order to determine the radial velocity in the gap, the specific volume must be known. The pressure p_s is measured. The temperature is computed as follows: The theoretical delivery head of the impeller up to the gap is $\mu_o \cdot \varrho_o \cdot u_2^2/g$; $\mu_o \cdot \varrho_o$ and ϱ_o respectively is still an unknown quantity but can be estimated very closely, as soon as the outlet triangle is approximately known. The temperature in the gap is therefore given by

$$T_s = T_o \left[1 + \frac{k-1}{k} \cdot \frac{H_o}{RT_o} \varrho_o\right]$$
$$= T_o \left[1 + (k-1) \mu_o \varrho_o M_o^2\right] \tag{7}$$

Here $M_o = u_2 / \sqrt{kg\, RT_o}$ is the Mach number. The specific volume in the gap amounts to $v_s = RT_s/p_s$ and the radial component of the outflow velocity is $c_{2r} = G \cdot v_s / \pi D_2 b_2$, and from this $\nu_{2r} = c_{2r}/u_2$ may be calculated.

The peripheral component of the delivery velocity of the impeller can be calculated from the power input. In the case of inflow without angular momentum $H_o = L/G$ and $\nu_{2u} = gH_o/u_2^2 = \mu_o$.

$\mu_o \varrho_o$ can now be determined exactly, it is hardly necessary to repeat the calculation for ν_{2r}.

The polytropic work of compression in the impeller is

$$H_s = RT_o \cdot \frac{n}{n-1} \left[\left(\frac{p_s}{p_o}\right)^{\frac{n-1}{n}} - 1\right] \tag{8}$$

from this we deduce $\mu_s = gH_s/u_2^2$ and from this $\eta_R = \mu_s / \mu_o \varrho_o$. Unfortunately, the polytropic coefficient n depends on η_R so that n must first be assumed

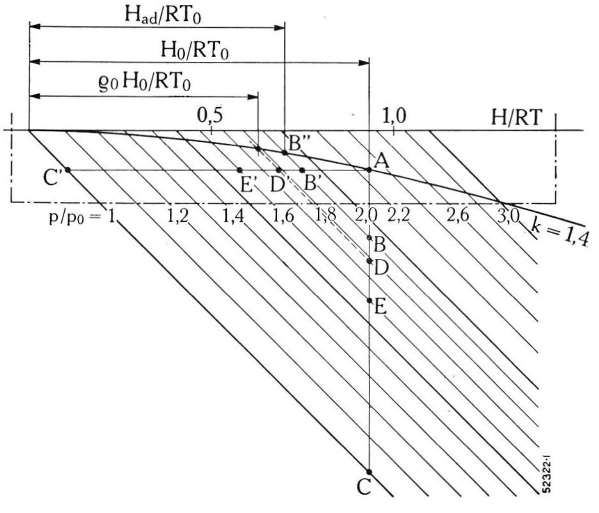


Fig. 3. — **Diagram to determine the efficiencies from the pressures measured.**

A. Theoretical pressure corresponding to the delivery head H_o.

B. Pressure measured.
Efficiency of blower:— η_{pol} = BC/AC = B'C'/AC'.

D. Theoretical pressure in gap corresponding to the delivery head $\varrho_o H_o$.

E. Pressure measured in gap.
Impeller efficiency:— η_{Rpol} = CE/CD = C'E'/C'D'.
Diffusor efficiency:— η_{Dpol} = BE/AD = B'E'/A'D'.

For practical purposes, only the part of the diagram above the dash-dot lines need be drawn. To make things clearer, only one line, for k = 1·4 is shown.

and then eventually corrected. This can be avoided, however. For the ideal comparative compressor we have:

$$H_2 = RT_o \cdot \frac{k}{k-1}\left[\left(\frac{p_2}{p_o}\right)^{\frac{k-1}{k}} - 1\right] \qquad (9)$$

where the ideal gap pressure $p_2 > p_s$. By dividing (8) by (9) we get

$$\frac{H_s}{H_2} = \eta_R = \frac{n}{n-1}\cdot\frac{k-1}{k}\,\frac{\left[\left(\frac{p_s}{p_o}\right)^{\frac{n-1}{n}} - 1\right]}{\left[\left(\frac{p_2}{p_o}\right)^{\frac{k-1}{k}} - 1\right]} \qquad (10)$$

As now $\quad \frac{n}{n-1}\cdot\frac{k-1}{k} = \eta \qquad (11)$

therefore

$$\frac{p_s}{p_o} = \left(\frac{p_2}{p_o}\right)^{\eta_R} \qquad (11)$$

or

$$\eta_R = \frac{\log (p_s/p_o)}{\log (p_2/p_o)} \qquad (12)$$

The ideal gap pressure ratio p_2/p_o can be deduced from equation (9)

$$\frac{p_2}{p_o} = \left(\frac{T_s}{T_o}\right)^{\frac{k}{k-1}} = \left[1 + \frac{k-1}{k}\cdot\frac{H_o}{RT_o}\cdot\varrho_o\right]^{\frac{k}{k-1}} =$$

$$\left[1 + (k-1)\,\mu_o\,\varrho_o\,M_o^2\right]^{\frac{k}{k-1}} \qquad (13)$$

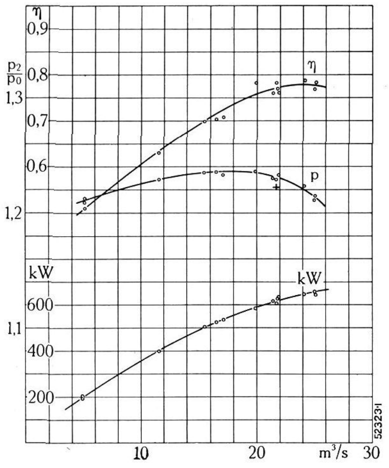


Fig. 4 a. — Characteristic of a single-stage blower.

kW = Power. η = Efficiency.
P = Pressure. + = Normal point.

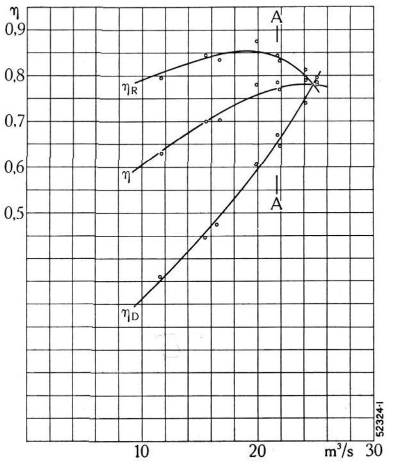


Fig. 4b. — Impeller and diffusor efficiencies.

η_R = Impeller efficiency. η = Blower efficiency.
η_D = Diffusor efficiency. A-A = Normal point.

The characteristics according to Fig. 4a seem quite normal. It is only after the impeller and diffusor efficiencies have been separated according to Fig. 4b that it is seen that the diffusor is much too big.

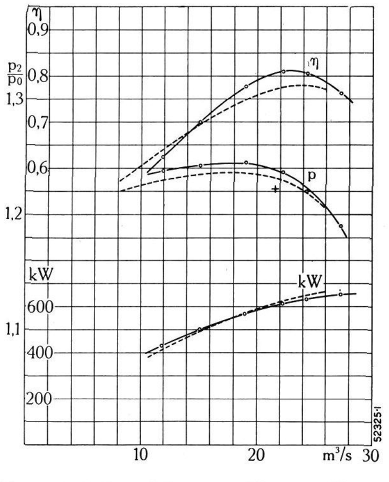


Fig. 5a. — Characteristics of the same blower with a smaller spiral housing.

kW = Power. η = Efficiency.
P = Pressure. + = Normal point.
------ = Curves for original spiral housing.

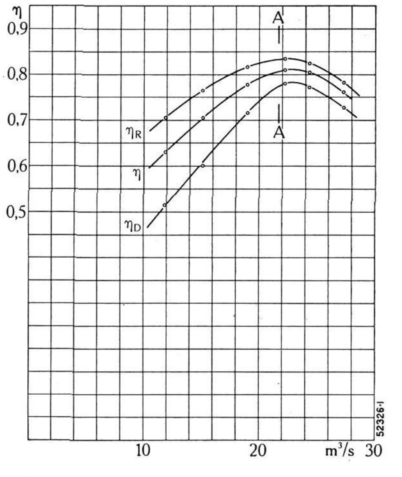


Fig. 5b. — Efficiencies of impeller and diffusor.

η_R = Efficiency of impeller. η = Total efficiency.
η_D = Efficiency of diffusor. A-A = Normal point.

The same blower represented as in Figs. 4a/b but with a smaller spiral housing. The diffusor is now of the right size. In Fig. 5a the curves from Fig. 4a are also shown, for comparison, as dash lines. The efficiency is increased by 3 % and the character of the pressure curve improved.

Analogously for the diffusor we get

$$\eta_D = \frac{\log (p/p_s)}{\log (p_{th}/p_s)} \qquad (14)$$

where p_{th} is the theoretical delivery head of the blower — that is at efficiency = 100 %. For the entire blower

$$\eta = \frac{\log (p/p_o)}{\log (p_{th}/p_o)} \qquad (15)$$

These expressions are so simple because the gap temperature T_s is the same for the ideal and for the real blower.

These formulæ can serve as basis for a very simple diagram.— The expression

$$f(x;\, k) = \frac{1}{k-1} \log_e \left(1 + \frac{k-1}{k} \cdot x\right) - x$$

can be represented graphically, as a function of x with k as parameter. For k = 1 (isothermal compression) f (x; 1) = 0. Further the lines $\log_e (p/p_o) - x$ are drawn in for a suitable series of values p/p_o.

The utilization of the diagram [1] is shown in Fig. 3, for the sake of simplicity with only one curve corresponding to k = 1·4. If the theoretical delivery head $H_o = L/G$ is known (L power input; G = weight delivered) then we have AC = $\log p_{th}/p_o$ in the diagram. The actual pressure ratio attained is plotted in B, then $\eta_{pol} = BC/AC$. The work of compression in the impeller is $\varrho_o H_o$ and, therefore, the length CD = $\log (p_2/p_o)$. The pressure ratio measured in the gap is plotted in E so that $\eta_{R\,pol} = CE/CD$. For the diffusor analogously $\eta_{D\,pol} = BE/AD$.

By an artifice the diagram can be made more handy. It is at once evident that AC = AC′ and AB = AB′. It is, therefore, more practical to draw a horizontal line through A to C′ and to plot the pressure really measured at B′. Then $\eta_{pol} = B'C'/AC'$; $\eta_{R\,pol} = C'E'/C'D'$ and $\eta_{D\,pol} = B'E'/AD'$. The whole lower part of the diagram can, thus, be eliminated. [2]

The illustrations Figs. 4 and 5 demonstrate the practical value of the separate determinations of the efficiencies. Fig. 4 a shows the pressure curve, power-input curve and efficiency curve of a blower; the character of these curves is quite normal. In Fig. 4 b, however, the efficiencies of impeller and diffusor are represented separately. These curves show clearly that the diffusor is much too big, because even with the biggest amount of air delivered the maximum value is not attained.

Therefore, the blower was fitted with a small diffusor and the success of this modification is shown by Fig. 5 a. The efficiency rose by 3 % and the character of the p—v curve was improved as well. Fig. 5 b shows that the efficiencies of the impeller and the diffusor now attain their maximum value simultaneously at the normal point, as indeed should be the case.

The separate measurements of the efficiencies of impeller and diffusor is, therefore, a very useful means towards further perfecting blowers and compressors.

(MS 770) *A. Meldahl. (Mo.)*

[1] This diagram is a modification of the polytropic entropy diagram for gases described in Mr. Zweifel's article on page 232 of this number.

[2] It should also be mentioned that the *adiabatic* efficiency can also be taken from the diagram. The adiabatic delivery head which would be required to compress the air to pressure B is, obviously, H_{ad}; therefore $\eta_{ad} = H_{ad}/H_0$.

THE SUPERCHARGING OF INTERNAL COMBUSTION ENGINE PLANTS DRIVEN BY PRODUCER WOOD GAS WITH SPECIAL REFERENCE TO MOTOR VEHICLES.

Decimal index 621.43.052.068.2 : 629.113.3

When an internal combustion engine is made over for producer wood gas drive, 40 % of the power is lost. This loss is made good again by means of turbo charging. Supercharging does not necessitate any alteration to the engine and only slight alterations to the producer.

THE increasing scarcity of liquid fuels as a result of the war has created a demand for substitutes. Producer gas from wood is the most important substitute for motor vehicles. Unfortunately, the petrol and Diesel engines driven by wood gas suffer a drop in power output which may attain 40 %. The chief cause of this is the low calorific value of wood gas as compared to petrol or to Diesel oil. Attemps had already been made to remedy this defect by supercharging the engine, a compressor being used to pump the gas under pressure into the engine. The impurities contained in wood gas, however, clogged the compressor rather rapidly so that this kind of service demanded constant attendance. Further, the mechanical drive of the compressor absorbed a not inconsiderable fraction of the power recuperated.

We, therefore, sought a new solution. Instead of only compressing the wood gas we put the producer itself under pressure. To generate the requisite pressure and also to deliver combustion air under the same pressure, a blower is used which is driven by a gas turbine supplied with exhaust gas from the engine. Thus, no

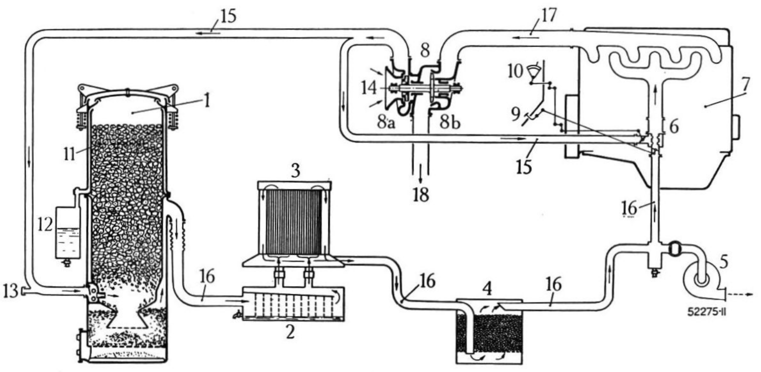


Fig. 1. — Diagram of a wood-gas producer plant for motor vehicles, with turbo charging.

1. Imbert producer.
2. Baffle-plate settling filter.
3. Cooler.
4. Fine-filter.
5. Starting fan.
6. Mixing nozzle for producer gas and air.
7. Motor.
8. Turbo charger.

8a. Blower.

8b. Gas turbine.

9. Gas pedal.
10. Air lever.
11. Wood load.
12. Seepage water trap.
13. Ignition aperture.
14. Air inlet to blower.
15. Combustion-air pipe.
16. Wood-gas pipe.
17. Exhaust gas pipe from engine.
18. Exhaust to atmosphere.

useful power is diverted to drive the blower, it only utilizes energy which would otherwise be lost to atmosphere in the engine exhaust. Fig. 1 shows the diagrammatic layout of a plant of this type. It should be noted that the charger is placed in front of the gas producer.

It is a well-known fact that the output of an internal combustion engine is increased by supercharging. The increase in power is due to the cylinder drawing in a bigger quantity of fuel-air mixture at each piston stroke and also to the work during the suction operation, which is otherwise considerable, being eliminated.

Supercharging has also special advantage in wood-gas producer drive. Engines, originally built for petrol drive for example, need not be made over for a higher compression ratio as is the case when non supercharged wood-gas producer drive is adopted, because the supercharging process itself generates the necessary higher pressure in question.

Further, the engine does not require any additional liquid fuel to make self-ignition possible, such as is needed for the engines working according to the Diesel process which have recently been put on the market.

In order to compensate to some extent for the drop in power of non supercharged wood-gas producer engines, it has been suggested that the engine be made to run at a higher speed. This is feasible with some engines but naturally shortens the useful life of the whole plant.

Thanks to the wide experience we had acquired in the field of supercharging, we did not expect to meet with any new difficulties in this new application of the process and, indeed, encountered none. However, it must be born in mind that the supercharging of the gas producer itself is a quite new departure. It would have been quite reasonable to expect some difficulties here as the course of the processes taking place in the gas producer is subjected to a new influence as is, indeed, the case in other supercharging processes. Firstly we thought it possible that a displacement of the zones, which play such an important part in gas producers, might occur. This would have meant that the apparatus built up till now for suction operation would have to be discarded or at least considerably altered.

Nothing of the kind occurred. The supercharging tests carried out with the wood-gas producer of the Imbert type gave satisfactory results from the beginning. The necessary alterations were limited to fastening on the charging cover of the producer by powerful springs so that the fitting of the cover on the producer housing is gas tight and, will only rise after an internal pressure of $0 \cdot 5$ kg/cm^2 has been exceeded, thus acting as a safety valve.

It is impossible to say definitely to-day to what extent a displacement of the gas generating and charcoal forming zones takes place. On the other hand, the consumption of charcoal which has to be replaced periodically in the annular space round the hearth seemed to be considerably reduced and hardly to attain 1 kg per 100 kg of wood consumed.

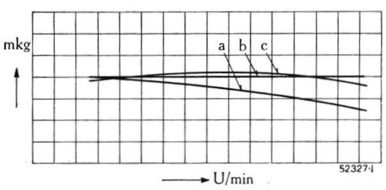


Fig. 2. — Characteristic of the torque in function of the speed.

a. On a Diesel driven Saurer road truck made over for wood-gas producer drive, not supercharged.

b. The same supercharged.

c. Diesel engine.

The supercharged wood-gas producer engine and the Diesel engine have torque curves which nearly coincide.

The measurement results attained with supercharging on the test bed are reproduced in Figs. 2 and 3. Fig. 2 shows the engine torque at various speeds, with and without supercharging. For purposes of comparison, the torque of a truck Diesel engine driven by gas oil is given. The considerable gain due to supercharging is obvious as well as the fact that the characteristic very nearly coincides with that of the Diesel engine.

As has already been mentioned, it is not necessary to make over the engine for a higher compression ratio. Fig. 3 confirms this. It shows the output of

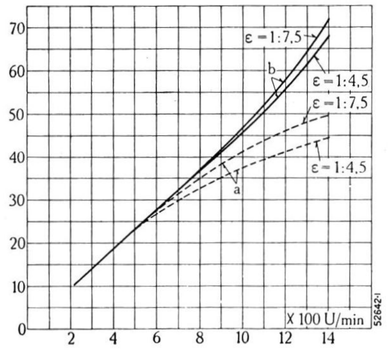


Fig. 3. — Power increase of a petrol engine with unchanged compression ratio 1 to 4·5 and of a petrol engine with increased compression ratio 1 to 7·5 in function of the speed, both driven by wood gas.

Ordinates: Output in H.P.

a. Without supercharging.

b. With supercharging.

In the supercharged engine, the increased compression ratio does not cause any essential increase in power, while it is appreciable if the engine is not supercharged.

two engines in function of the speed, one of which has a compression ratio of 1 to 4·5, as is common practice in petrol engines, and the other the higher compression ratio of 1 to 7·5. Without supercharging there is a very marked difference in the outputs while with supercharging this difference is small, especially when the supplementary power available due to supercharging, is taken into consideration.

The results recorded on the test bed are generally valid for practical service. Thanks to supercharging, the entire engine output is again made available and thus restores its former power capacity to the vehicle. It is characteristic that, for example, the same vehicle under identical conditions can take a grade in higher gear when supercharged than without supercharging. As the engine no longer draws in fuel and gas directly but gets them pumped into it by a blower which must first be brought up to speed by a gas turbine, the technic of driving is subjected to a slight modification. Thus, for example, it is better to change gears a little earlier before negotiating hills in order to give the blower time to accelerate, so that the full amount of air and gas may be available when wanted. Further, the accumulation of gas resulting from the higher pressure in the producer, calls for more pronounced throttling to bring down the engine output than under ordinary suction drive. However, these are small matters with which a driver rapidly acquaints himself. Further we are at present developing regulating devices the object of which is to simplify control.

Fig. 4 shows a turbo-charger for 40 to 150 brake H. P. plants. Its biggest diameter is 280 mm, length 350 mm and it weighs about 35 kg. It can be built into any car without difficulty. Having its own lubricating system, it is independent of that of the car. Fig. 5 shows how a charging set is mounted on the frame of a "Berna" road truck.

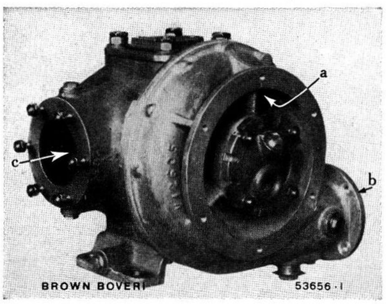


Fig. 4. — Charging blower driven by an exhaust-gas turbine, Type VT 100.

Seen from the air inlet side.

a. Fresh-air inlet.

b. Charging air outlet.

c. Exhaust-gas inlet to the gas turbine.

This set is for plants of 40 to 150 H. P.

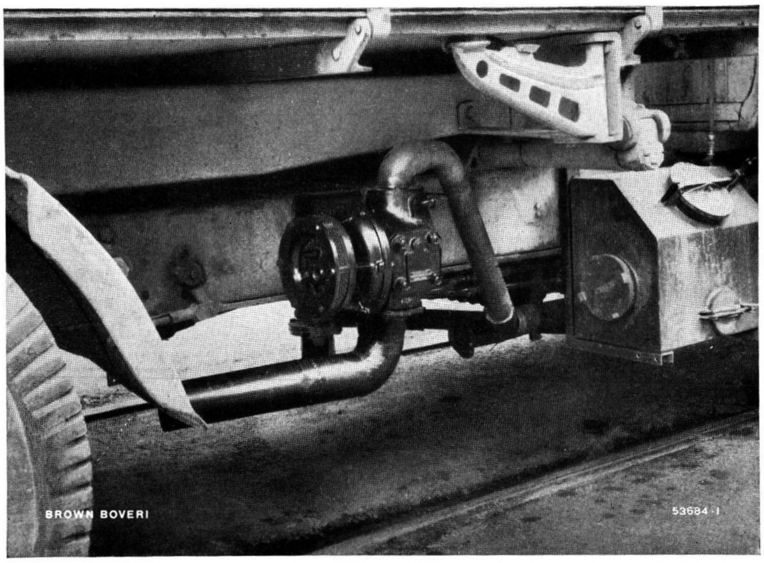


Fig. 5. — Turbo charger built on to a "Berna" road truck, with Imbert wood-gas producer.

(MS 763) *W. Meyer. (Mo.)*

SOME COMMENTS ON THE PROPERTIES OF MATERIALS USED IN THE MANUFACTURE OF THE GAS TURBINE.

Decimal index 621.438.0023

Attention is drawn to the importance of recent investigations carried out on constructional materials with reference to the development of the gas turbine. It is of the greatest importance that steels of a high creep limit should be available and that designers should be familiar with their behaviour at high temperatures. Increasing importance is also attached, in this field, to length of life or admissible time during which the material can be subjected to a stress.

AFTER decades of effort, the gas turbine has, at last, become a practical proposition. Apart from the great progress accomplished in the study of the flow of gases, this realization of a long sought for aim is certainly due to the advances made in the manufacture of heat-resisting steels. This is best demonstrated by the recording, year by year, of the values considered as the highest attainable ones in creep-limit tests (Fig. 1). For these values, not only the allowable stressing due to loading and the elastic elongation resulting therefrom, as defined by the Hook law, have to be considered, as was the case formerly, but paramount importance is attached, to-day, to the length of time during which the material is subjected to the stress at high temperatures. The elongation to be expected after a certain time, at a given temperature and under a given load, the so-called creep is determined by long-time tests which allow sufficiently reliable conclusions to be drawn as to what the character of the subsequent[1] elongation will be (Fig. 2a and b).

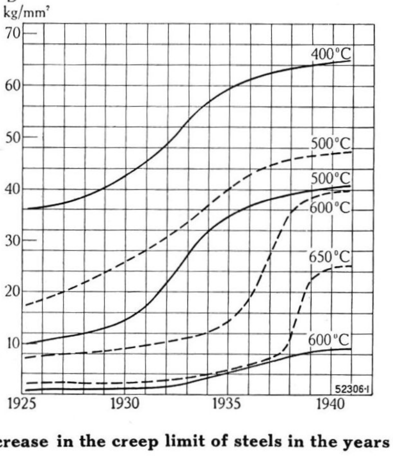


Fig. 1. — Increase in the creep limit of steels in the years 1925—1941.

——— low-alloyed martensitic steels.
- - - - high-alloyed austenitic steels.
D. Maximum values of the creep limit measured.

The rapid rise in the years 1935—39 is noticeable. It has allowed of building really serviceable gas turbines. No prediction can be made as to the character of the development in future years.

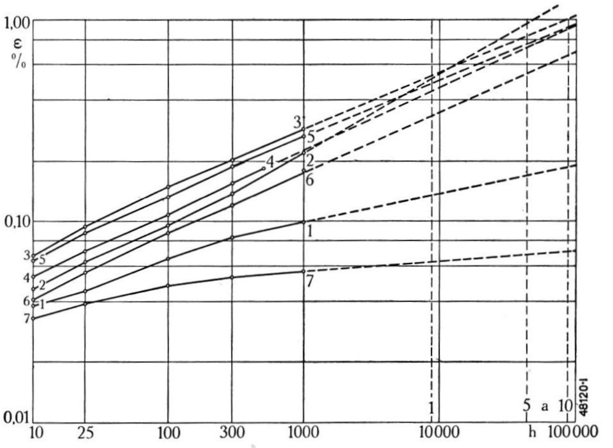


Fig. 2a. — Characteristic of the elongation in long-time tests at high temperatures for different constructional steels.

ε. Elongation.
h. Testing time in hours.
a. Testing time in years (extrapolated).
Steel 1—7 see Table I.

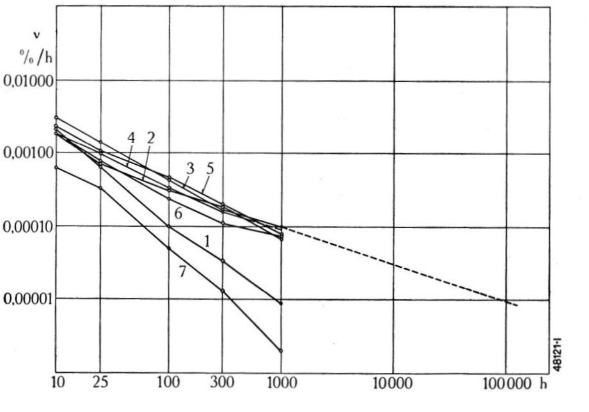


Fig. 2b. — Characteristic of the creep rate in long-time tests at high temperature for different constructional steels.

v. Rate of creep.
h. Testing time in hours.
Steel 1—7 Table I.

In the practical application of the creep-limit figures supplied by creep-limit tests another observation recorded gains in importance, namely the creep re-

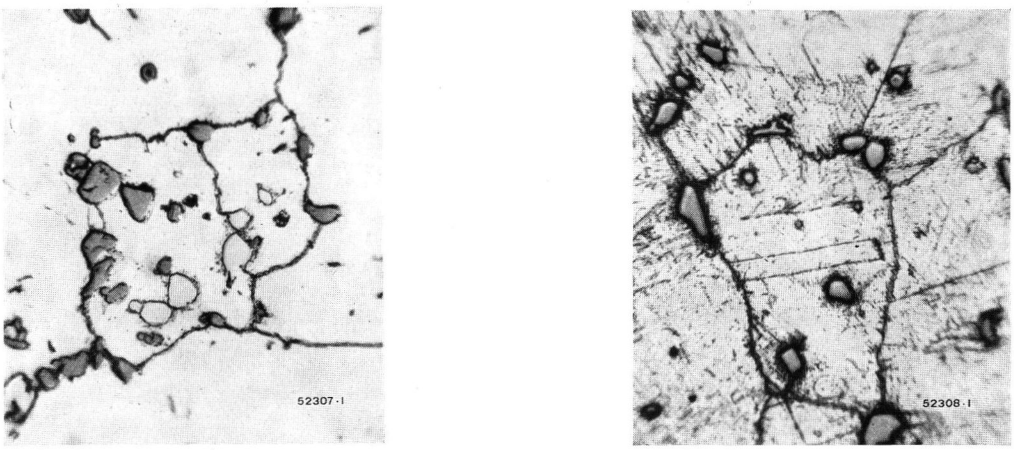
[1] See The Brown Boveri Review, 1938, page 247.

TABLE I.

Data on the long-time tests (see Figs. 2a, 2b).

No.	Steel	Temperature °C	C %	Cr %	Ni %	Mo %	W %	Continuous stress kg/mm²	Elongation measured after 1000 h %	Assumed elongation after 10 years %
1	Non-alloyed C steel . . .	400	0·25	—	—	—	—	22	0·100	0·190
2	„ „	600	0·25	—	—	—	—	2	0·230	1·320
3	Cr Mo constructional steel	400	0·3	0·8	—	0·5	—	60	0·290	1·050
4	„ „	500	0·3	0·8	—	0·5	—	11	0·230	0·960
5	Mo cast steel	500	0·2	+0·2	Cu	0·4	—	14	0·265	0·960
6	Stainless Cr Mo steel . .	500	0·5	14·7	—	1·2	—	28	0·181	0·700
7	Austenitic Cr Ni steel . .	500	0·11	17·4	8	—	1	35	0·057	0·072

covery. Turbo machinery is not subjected to the highest stressing and temperatures constantly, or all the time they run; there are periods of relief on unloading which reduce in a most welcome manner the amount of the total elongation. In the course of prolonged heating, changes in the structure of the material take place, translation movements in the cleavage surfaces (Fig. 3) and other processes may accelerate the creep rate while periods of unloading may retard these effects or compensate them to a certain extent.



a. As received. b. After 48 h, 650° C., 20 kg/mm².

Figs. 3a and 3b. — Translation movements in the cleavage surfaces and grain boundaries of a high-alloy steel as a result of continuous heat and stress.

(Magnified 1100 ×)

These translation movements caused this high-alloy Cr Ni steel to become completely brittle.

It is well known that the time factor plays a part in rapidly alternating fatigue stressing or rather that a given material subjected to a determined stress in excess of the fatigue limit will only stand up to a given number of stress cycles after which it breaks. This led to the so-termed time limited-endurance which indicates how severely a part can be stressed, for a length of life determined in advance.

Time-limited endurance and creep limit with or without regard to creep recovery will serve together as basis of calculations when it is a case of attaining very light weights. A case in point is the exhaust-gas driven charging set of aeroplane engines. Its length of life is that of the aeroplane motor. This length of life is influenced both by temperature and by fatigue and is determined by admissible rate of creep and time-limited endurance.

These summary remarks show that for materials used in gas turbine construction there are still interesting problems to solve. Here is a field of research in which there is still much to be done.

(MS 773) *H. Zschokke. (Mo.)*

TEST BED FOR EXHAUST-GAS TURBO-CHARGING SETS.

Decimal index 621.006.2 : 621.43.052.068.2

Exhaust-gas turbo-charging sets are tested before they leave the shops. A special test bed has been made for this purpose with the object of carrying out tests under conditions which are as alike those met with in practice as is possible. The test bed is described in this article and attention drawn to the importance to be attached to the way the tests are made, with regard to the development of economical and reliable charging sets.

UP till to-day, we have built more than 1500 exhaust-gas turbo-charging sets for supercharging Diesel-engines of outputs ranging from 80 to 5000 H. P. Before the charging set is allowed to leave our shops, it is subjected to tests. It would seem obvious that a set to be combined with a Diesel engine should be tested along with an engine of the same type. However this method is not feasible from the practical point of view, because of the variety of sizes to which the charging sets are built. Therefore, in the first stage of development, steam was resorted to as a testing medium. But steam proves too expensive because the heat drops which the exhaust-gas turbines are designed for are always very small while the quantities of exhaust gases they are built to handle are often very considerable. The heat drop only amounts to 10—15 kcal/kg corresponding to 0·3 — 0·4 kg/cm² superpressure while the amount of exhaust gas attains 3·6 to 4 kg/h for each H. P. of Diesel-engine output to which figure a certain percentage of scavenging air may have to be added. Further, the steam for the test run must be considerably overheated in order to attain the temperature of the exhaust gases which the set will have to deal with and which, as is well known, reach 500—600°C in four-stroke Diesel engines and even as much as 1000°C in aeroplane engines.

For this reason turbo-chargers are now tested with the same driving medium used in practice namely exhaust gas. This exhaust gas is generated in a combustion chamber by burning gas oil and the temperature it is desired to attain is reached by mixing air with it (Fig. 1). According to what data it is desired to obtain, there are two ways of proceeding:—

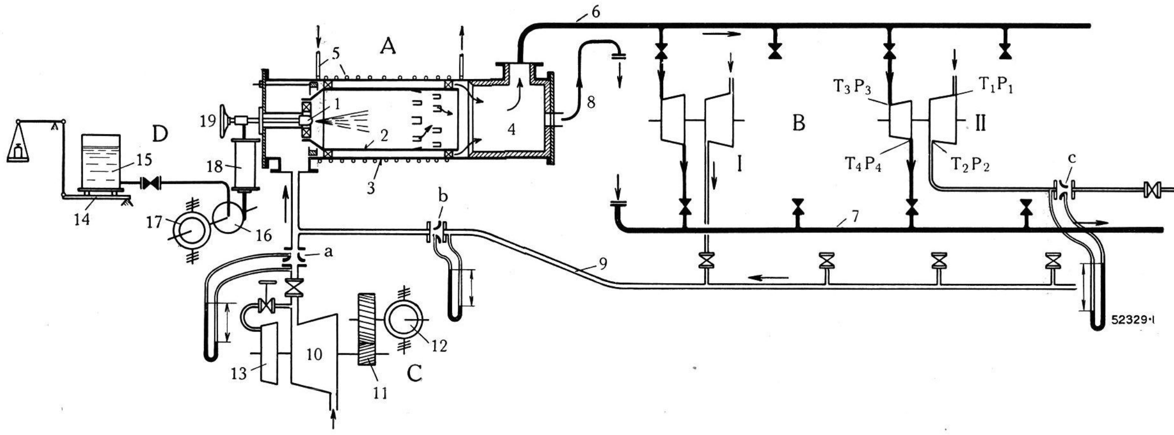


Fig. 1. — Layout diagram of the test bed for charging sets.

According to the object in view there are two testing methods available.

I. The charging set to be tested in closed-circuit operation. The air necessary to produce the driving gas is delivered by the blower itself.

II. The charging set to be tested in open circuit or operation with a source of external compressed air. The air necessary to produce the driving gas is delivered by an auxiliary blower.

A. **Combustion chamber.**
1. Burner.
2. Heat-proof wall of combustion chamber.
3. Pressure-proof wall of external housing.
4. Mixing chamber.
5. Cooling coils.

B. **Test bed.**
6. Driving-gas (exhaust-gas) inlet pipe.
7. Pipe to chimney.
8. Separate connection for driving gas of exceptionally high temperature.
9. Return pipe for charging air.

C. **Auxiliary set.**
10. Blower.
11. Reduction gear.
12. Motor.
13. Recuperation turbine.

D. **Fuel supply.**
14. Scales.
15. Fuel tank.
16. Fuel pump.
17. Motor.
18. Filter.
19. Fuel governor.

a, b, c. VDI standard nozzles with mercury columns.

If the first object is to ascertain the overall efficiency of the set, the air delivered by the charging set itself is taken as combustion and mixing air, as takes place in practice. Thus, the blower delivers air to the combustion chamber and the driving medium of the gas turbine is the air the charging set generates itself to which must be added that amount of fuel requisite to heat the air (therefore, this is a closed circuit or closed cycle process).

If, however, exhaustive testing of the charging set is aimed at as, for example, the determination of the whole characteristic curves, the driving medium is generated by means of a separate blower and the compressed air delivered by the blower of the set under test is exhausted to atmosphere (this is termed open circuit or auxiliary-air drive).

Apart from the most varied measurements thus made possible, both these methods allow of exact observation of the mechanical behaviour of the blower especially as regards the influences of temperature. This is very important for sets such as are used for very highly-stressed Diesel engines and, above all, for aeroplane engines. The necessary clearance for the blades as well as deformations can be checked, further the behaviour of bearings and efficacity of lubrication etc. can be tested.

It is true that the two testing methods just described still differ from real service conditions in conjunction with Diesel engines in so far as they are carried out at a pressure which does not vary during the duration of the test. As is well known, increasing use is being made to-day in the supercharging of Diesel engines, of the so-termed impact process, this means that the turbo-charger is placed as close to the Diesel-engine cylinders as possible while the exhaust-gas duct and gas-turbine nozzles are so dimensioned that the instantaneous discharge of exhaust-gases at the opening of the exhaust valve causes a considerable pressure rise to build up in front of the gas-turbine nozzles, which is followed by a drop in pressure during the exhaust stroke of the Diesel engine piston. These pressure fluctuations cannot be reproduced on the test bed. The test results attained at constant exhaust-gas and compression pressures are, however, quite sufficient in order to judge of the qualities of the charging set when it has to work with a Diesel engine the working characteristics of which are known.

The investigations carried out on the test bed naturally do not prevent the charging set from being tried out again exhaustively after it has been mounted with the Diesel it is to work with. These further tests are very essential when dealing with a new type of charger or with a type of Diesel engine which has never been supercharged before. For standard-production charging sets, the test-bed test sufficies entirely, as has been said, it is even usually possible to do without the test with hot exhaust gases and only to test with cold compressed air, in order to determine if the mechanical construction of the set is flawless.

In any case, the numerous testing possibilities offered show that everything is done to assure that only sets of as perfect workmanship as possible leave our shops and are delivered to engine manufacturers for building on to their machines.

As regards the test layout Fig. 1, it should be added that the combustion chamber is similiar to that of independent gas turbine plants. It consists of the interior combustion chamber, of a cooling space surrounding it and of the mixing chamber connected to it. The liquid fuel (gas oil) is burnt in the combustion chamber proper with little excess air. The major portion of the air flows through the gap between the combustion chamber wall, of very heat-proof metal sheeting, and the pressure-proof external wall, at high velocity, so that the walls are effectively cooled. In the mixing chamber the real driving gas is finally formed, the desired temperature being reached by the proper mixture of combustion gas and cooling air.

As especially high temperatures up to 1000^0 C are required for the testing of aeroplane engines, for example, the mixing room is provided with a lining of fire-proof brickwork, while tube coils through which water flows are wound round the cover of the combustion chamber: the duct leading to the charging set under test is built specially of very heat-proof metal sheeting. It may well happen that this duct gets heated up to a red glow during tests. An ordinary pressure-type atomizer is used for introducing the fuel; the oil consumption is measured on a scale. A small worm-type pump raises the oil to the atomizing pressure while a filter retains impurities. VDI standard nozzles are used to measure the amount of air delivered while all temperatures are measured by precision thermometers.

The efficiencies, which are the most important factors to be ascertained, are given by the quantity and temperature measurements as follows:

If

G_L · Weight of air delivered by blower

c_{pL} · Specific heat of air under constant pressure

$\Delta t_{ad\,L}$ · Temperature rise under adiabatic compression from P_1 to P_2

G_A · Weight of driving or exhaust gas used in the gas turbine

c_{pA} · Specific heat of driving gas under constant pressure

$\Delta t_{ad\,A}$ · Temperature drop under adiabatic expansion from P_3 to P_4

then the overall efficiency of the charging set is given by

$$\eta_{tot} = \frac{G_L \; c_{pL} \Delta \; t_{ad\,L}}{G_A \; c_{pA} \Delta \; t_{ad\,A}}$$

This value is usually quite sufficient for judging of the quality of the set. It also takes into account the mechanical losses.

If it be desired to ascertain the separate efficiencies of blower and turbine, it is necessary to measure the temperatures of the gas and of the air at the inlet and outlet. The thermodynamic efficiencies are then, as is known, the quotients of the adiabatic and real temperature difference

$$\eta_G = \frac{\Delta t_{ad\,L}}{T_2 - T_1} \quad \eta_T = \frac{T_3 - T_4}{\Delta t_{ad\,A}} \text{ and } \eta_m = \frac{\eta_{tot}}{\eta_G \times \eta_T}$$

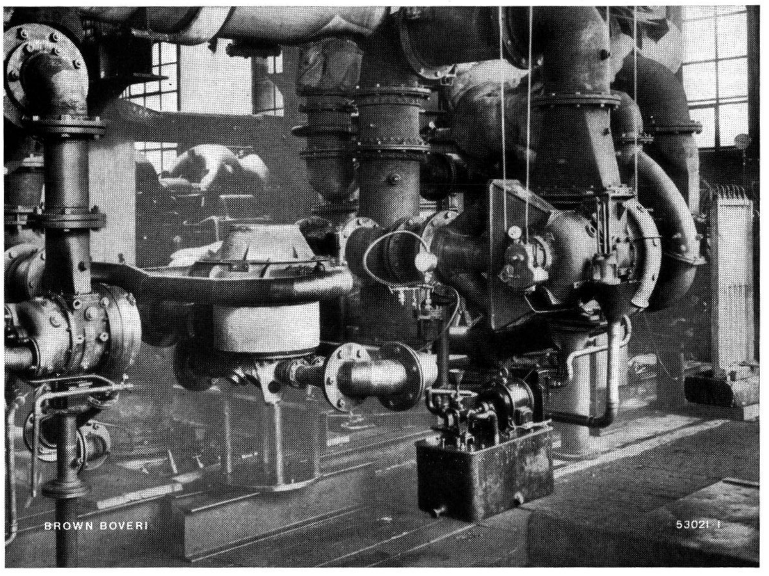


Fig. 2. — View of test bed for turbo-charging sets showing four sets connected up for simultaneous testing

From left to right: Charging set for 300 H. P. Diesel engine.
Charging set for 600 H. P. Diesel engine (traction engine).
Charging set for 1000 H. P. Diesel engine.
Charging set for special tests.

here η_G = thermal efficiency of blower, η_T = thermal efficiency of turbine and η_m = mechanical efficiency. In ordinary tests, as has already been mentioned, these factors are not determined and this the more so as temperature measurements are never very precise. For the turbo-charging sets of aeroplane engines in which a part of the air is led directly to the turbine wheel as cooling air, the above equations are no longer valid.

The test bed is laid out for the simultaneous testing of several sets. Fig. 2 shows 4 sets of different designs connected up to the testing equipment. The board with the mercury columns will be noticed on the extreme right, by means of which the pressures at the different points of interest on the air and gas circuits are measured.

When, on the one hand, we decided on the building of this test bed some time ago and when, on the other, despite all the experiences already gained, we still go on subjecting all charging sets to more or less exhaustive tests, our reason for so doing is to make sure that clients are getting charging sets which are as perfect as it is possible to make them.

(MS 764) *E. Erni. (Mo.)*

THEORETICAL CONSIDERATIONS ON ALTITUDE SUPERCHARGING OF AEROPLANE ENGINES BY MEANS OF CHARGING BLOWERS DRIVEN BY EXHAUST-GAS TURBINES.

Decimal index 621.43.018.73 : 621.43.052.068.2

Curve sheets are established by calculation which permit of determining the energy balance sheet of the charging set under certain assumptions, in a very simple way. In particular they allow of following the changes which accompany an increasing degree of supercharging.

It is found that at about 5000 m above sea level, the turbine output available is at a minimum. Under certain conditions, more than one stable running point of the charging set is possible.

An illustration is given of a charging set for an aeroplane engine built by Brown Boveri and shown on the test bed when operating at a temperature of 1000° C (1800° F).

ALTITUDE supercharging of aeroplane engines by means of turbo-blowers driven by exhaust-gas turbines differs from ordinary charging of internal combustion engines because the process is much more accentuated. Whilst the output of stationary engines can be increased by 50 %, that of rail car engines by 80 % by supercharging, an aeroplane engine to give its full output at 12,000 m altitude has to be supercharged so as to give four times its output without supercharging. Thus, altitude supercharging offers certain peculiarities.

In the following calculations, it is assumed that the gas turbine operates to the constant-pressure process. In reality efforts always tend to turn the energy of the exhaust impacts to useful account. In altitude charging, however, the benefit from these impacts gets gradually smaller, so that calculations carried out on the assumption of constant pressure drive produce a suitable basis to work upon.

For the purpose of calculations which can be generally applied, ratio figures and not dimensions are introduced everywhere.

Thus:—

G_T = Weight of gas supplied to the turbine.
G_V = Weight of air supplied to the blower.
G_o = Weight of gas supplied to the engine *without* charging.
p_T = Pressure at turbine inlet branch.
p_V = Pressure at blower discharge branch.
p_o = Atmospheric pressure (varying according to altitude).
T_T = Temperature at turbine inlet branch.
T_V = Temperature at blower outlet branch.
T_o = Atmospheric temperature.
L_T = Energy of adiabatic expansion.
L_V = Energy of adiabatic compression.

η_T = Adiabatic efficiency of the turbine.
η_V = Adiabatic efficiency of the blower.
f_T = Section of turbine nozzle.
f_o = "Equivalent nozzle section" of the engine.

Further:—

$x = G_V/G_o$ = Ratio of charging process.
$\pi_T = p_T/p_o$ = Pressure ratio of turbine.
$\pi_V = p_V/p_o$ = Pressure ratio of blower.
$\tau_T = T_T/T_o$ = Temperature ratio of turbine.
$\tau_V = T_V/T_o$ = Temperature ratio of blower.
$\lambda_T = L_T/RT_T$ = Available power in exhaust gas.
$\lambda_V = L_V/RT_o$ = Power necessary for compression.
$\varphi_T = f_T/f_o$ = Section ratio of the turbine nozzle.

Three alternatives can be distinguished in the supercharging process:—

I. The air is cooled down again to its initial temperature after passing through the blower.

II. The air is not cooled down.
 a) The motor corresponds to a fixed diaphragm.
 b) The motor absorbs a constant volume of air.

The amount of air which is taken in by the motor is then:—

$$\begin{aligned} &\text{Case I:} && G_V = G_o \cdot p_V/p_o \\ &\text{Case IIa:} && G_V = G_o \cdot p_V/p_o \cdot \sqrt{T_o/T_V} \\ &\text{Case IIb:} && G_V = G_o \cdot p_V/p_o \cdot \sqrt{T_o/T_V} \end{aligned} \tag{1}$$

When the compression is *polytropic* we get:—

$$T_V/T_o = (p_V/p_o)^{(n-1)/n} \tag{2}$$

$$n = \text{Polytropic coefficient} = \frac{k\,\eta_{pol}}{k\,\eta_{pol} - (k-1)}$$

hence:—

$$\begin{aligned} &\text{Case I:} && \pi_V = x \\ &\text{Case IIa:} && \pi_V = x^{2n\,(n+1)} \\ &\text{Case IIb:} && \pi_V = x^n \end{aligned} \tag{3}$$

The *adiabatic* work of compression is

$$L_V = RT_o \frac{k}{k-1}\left[\left(\frac{p_V}{p_o}\right)^{\frac{k-1}{k}} - 1\right] \equiv RT_o \cdot \lambda_V \tag{4}$$

hence

$$\lambda_V = \frac{k}{k-1}\left(\pi_V^{\frac{k-1}{k}} - 1\right) \tag{5}$$

$$\begin{aligned} &\text{or Case I:} && \lambda_V = \frac{k}{k-1}\left(x^{\frac{k-1}{k}} - 1\right) \\ &\text{Case IIa:} && \lambda_V = \frac{k}{k-1}\left(x^{\frac{2n}{n+1}\cdot\frac{k-1}{k}} - 1\right) \\ &\text{Case IIb:} && \lambda_V = \frac{k}{k-1}\left(x^{\frac{n(k-1)}{k}} - 1\right) \end{aligned} \tag{6}$$

Above critical pressure ratio that is to say when the velocity in the nozzles of the turbine is equal to that of sound:—

$$G_T = f_T \frac{p_T}{\sqrt{RT_T}} \sqrt{kg\left[\frac{2}{k+1}\right]^{\frac{k+1}{k-1}}} \tag{7}$$

If we *define* the "equivalent nozzle section" of the engine as:—

$$f_o = G_o \frac{\sqrt{RT_o}}{p_o} \sqrt{\frac{1}{kg}\left[\frac{k+1}{2}\right]^{\frac{k+1}{k-1}}} \tag{8}$$

(f_o is the section of a nozzle through which G_o would flow at the velocity of sound when the pressure at the nozzle inlet is p_o) we get the equation

$$\pi_T = \frac{\sqrt{\tau_T}}{\varphi_T} \cdot x \tag{9}$$

above critical pressure ratio.

Below critical pressure ratio de Saint Venant's formula is valid

$$G_T = f_T \frac{p_T}{\sqrt{RT_T}} \sqrt{2g\frac{k}{k-1}\left[\left(\frac{p_o}{p_T}\right)^{\frac{2}{k}} - \left(\frac{p_o}{p_T}\right)^{\frac{k+1}{k}}\right]} \tag{10}$$

whence after calculation

$$x = \frac{\varphi_T}{\sqrt{\tau_T}} \sqrt{\frac{2}{k}\left(\frac{k+1}{2}\right)^{\frac{k+1}{k-1}}} \times \\ \times \sqrt{\frac{k}{k-1} \cdot \left[\left(\pi_T^{\frac{k-1}{k}} - 1\right) \cdot \pi_T^{\frac{k-1}{k}}\right]} \tag{11}$$

and solving for π_T

$$\pi_T = \frac{\sqrt{\tau_T}}{\varphi} \cdot \pi(x) \tag{12}$$

below critical pressure ratio.

In both cases:—

$$L_T = RT_T \cdot \frac{k}{k-1} \frac{\left(\pi_T^{\frac{k-1}{k}} - 1\right)}{\pi_T^{\frac{k-1}{k}}} \equiv RT_T\lambda_T \tag{13}$$

i. e.

$$\lambda_T = \frac{k}{k-1} \frac{\left(\pi_T^{\frac{k-1}{k}} - 1\right)}{\pi_T^{\frac{k-1}{k}}} \tag{14}$$

Operation is only possible when the turbine output is bigger or at least equal to the power absorbed by the compressor, i. e.:—

$$G_T L_T \cdot \eta_T \geqq G_V L_V/\eta_V \tag{15}$$

or

$$\eta_{ad} \cdot G_T R_T T_T \lambda_T \geqq G_V R_V T_o \lambda_V \tag{16}$$

here $\eta_{ad} = \eta_T \cdot \eta_V$ is the adiabatic efficiency of the charging set.

As $G_T R_T$ is practically equal to $G_V R_V$:—

$$\eta_{ad} = \tau_T \cdot \lambda_T \geqq \lambda_V \tag{17}$$

In the curve sheets, Figs. 1—3, the curves λ_V and π_V are calculated as functions for x for cases I, IIa

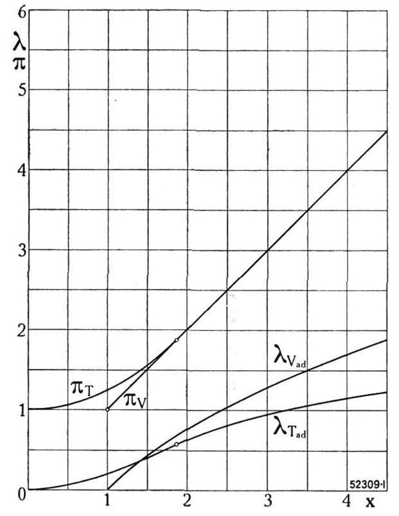


Fig. 1. — Diagram for the calculation of the altitude supercharging of an internal combustion engine.

x. Charging ratio.
λ_T. Available power in exhaust gas.
λ_V. Power for compression.
π_T. Pressure ratio of exhaust gas turbine.
π_V. Pressure ratio of blower.

Assumptions:—
The air is **cooled down to its initial temperature** after passing through the blower.

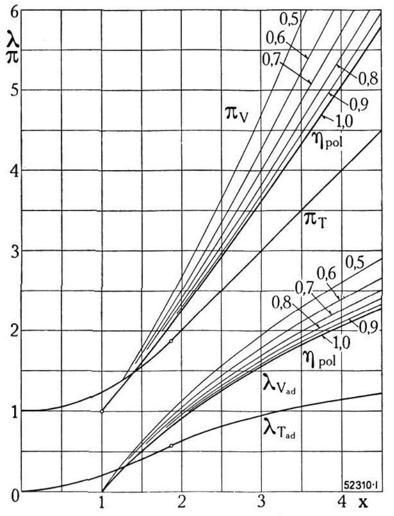


Fig. 2. — Diagram for the calculation of the altitude supercharging of an internal combustion engine.

x. Charging ratio.
λ_T. Available power in exhaust gas.
λ_V. Power for compression.
π_T. Pressure ratio of exhaust gas turbine.
π_V. Pressure ratio of blower.
η_{pol}. Polytropic efficiency of blower.

Assumptions:—
1. After compression the air is **not cooled down.**
2. The air consumption of the engine **corresponds to a fixed diaphragm.**

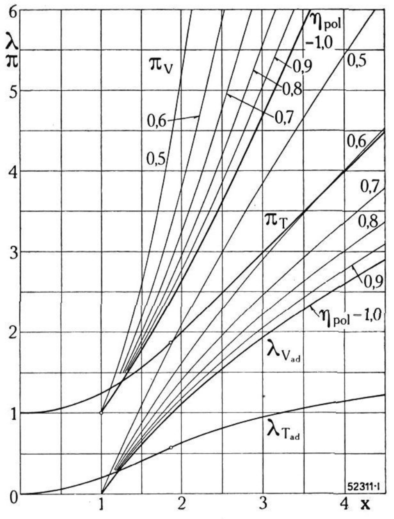


Fig. 3. — Diagram for the calculation of the altitude supercharging of an internal combustion engine.

x. Charging ratio.
λ_T. Available power in exhaust gas.
λ_V. Power for compression.
π_T. Pressure ratio of exhaust gas turbine.
π_V. Pressure ratio of blower.
η_{pol}. Polytropic efficiency of blower.

Assumptions:—
1. After compression the air is **not cooled down.**
2. The engine draws in a **constant volume** of air.

and IIb, further the curve λ_T and π_T for $\varphi_T = 1$ and $\tau_T = 1$. In order, now, to make the comparison according to equation 17, firstly curve λ_T must be stretched in the direction of the λ axis in the proportion $\eta_{ad} \cdot \tau_T$. Here, the efficiency for each point and, in the case of altitude flying, the external temperature T_o can be set in. From equations 9 and 12, together with 14, it may, on the other hand, be seen that the λ_T curve must, further, be stretched in the direction of the x axis in the ratio $\varphi_T/\sqrt{\tau_T}$. The deformed λ_T curve thus obtained must then correspond to equation 17, if operation is to be possible. From this, the value φ_T can be determined, that is to say the maximum admissible nozzle section.

Once φ_T has been determined in this way, the pressures before and after the engine can be compared. To do so, the curve π_T must also be stretched in the direction of the x axis, according to equations 9 and 12, in the same ratio $\varphi_T/\sqrt{\tau_T}$. According to how big φ_T had to be chosen with regard to equation 17, π_T is bigger or smaller than π_V.

It is quite obvious that a scavenging of the engine with excess air will only be possible if $\pi_T < \pi_V$. A glance at the curves shows that, under all circumstances, this is in any case only possible from a certain amount of charging upwards, when the nozzles are as small as it is necessary to make them for high altitude charging.

For about 1·8 times supercharging, the surplus of turbine power available is at a minimum. This corresponds to about 5000 m altitude above sea level, when the uncharged rated output at ground level is to be maintained. At higher altitudes, the surplus increases steadily in theory, but, in practice, the drop in the overall efficiency of the set limits charging, this in so far as the maximum allowable speed of the charging set has not already imposed a limit to the charging capacity of the set.

If the efficiency of the charging set decreases with *increasing* supercharging, it is possible, under certain conditions, that the distorted λ_T curve may cut the λ_V curve in *three* points. The lowest and highest of these points of intersection correspond to stable running conditions while the middle one corresponds to unstable conditions.

If the set is working at a great altitude, and at the highest of the above mentioned points, and if the engine is momentarily throttled, it may happen that the charging set passes through the unstable point and drops to the lowest point. If gas is again given to the engine, the set continues nevertheless to run on this lowest point and it is no longer possible to attain the earlier operating point without taking other measures.

This can be remedied by a transitory speed increase of the engine. This increases the weight of air G_o it draws in without supercharging and also f_o according to equation 8, so that φ_T drops. The increase in the speed of the engine acts exactly as would a closing of the turbine nozzles:— it increases the output of the turbine.

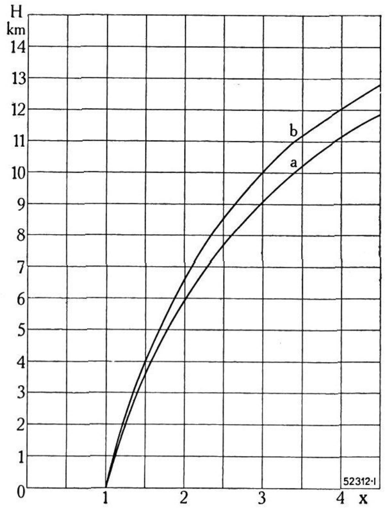
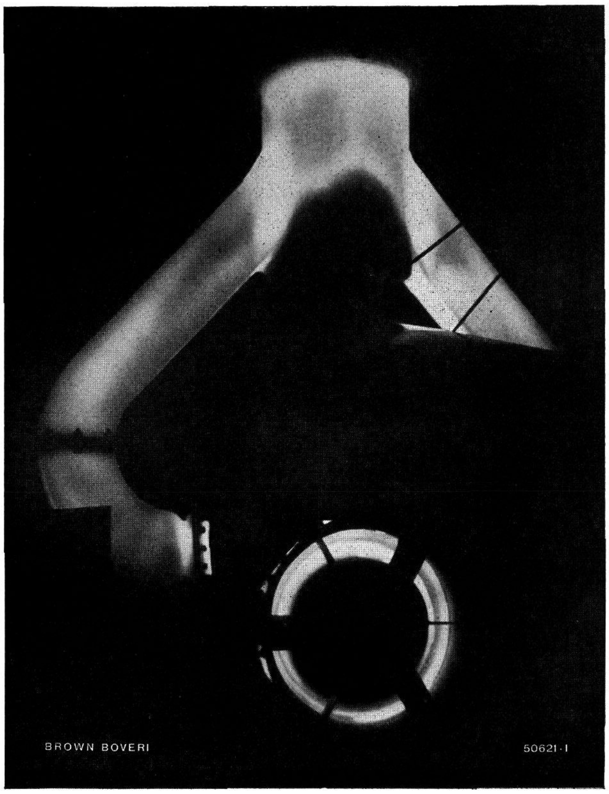


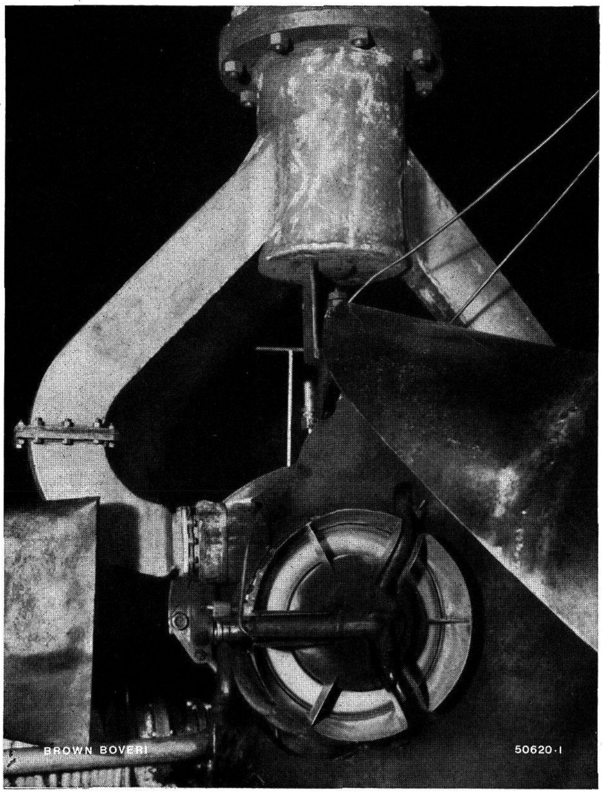
Fig. 4. — Necessary supercharging in order to obtain full-rated engine output at various altitudes.

H. Flying altitude in km.
x. Necessary supercharging.
a. The air consumption of the engine corresponds to a fixed diaphragm.
b. The motor takes a constant volume of air.
Curves calculated according to the International Normal Atmosphere rating.

In aeroplanes with adjustable propellers, this can be obtained by a momentary reduction of the propeller pitch untill full supercharging has again been arrived at. In a modern aeroplane with adjustable propeller pitch, the experience which Tomlinson went through in his high altitude flight in America will not be repeated (Report of the Lilienthal Society, Berlin of 13th Oct. 1938) according to which after a momentary throttling of the engine the charging set simply ceased to work forcing Tomlinson to drop until the set ran up to speed again automatically at some lower altitude.

The supercharging of aeroplane engines which is a factor of increasing importance to-day on account of the flights at very high altitudes which are being made has been studied by us not only theoretically but also practically. Many years ago, the first charging blowers for aeroplanes driven by exhaust-gas turbines were built and tested on a special test bed built for the purpose. In order to reproduce the high temperatures





Figs. 5 and 6. — Charging set for an aeroplane engine. The set comprises a blower and an exhaust-gas turbine, built for a gas temperature of 1000 ° C (about 1800 ° F). The illustrations show the set in continuous service at rated gas temperature and full speed of 30,000 r. p. m.

The night photograph shows the turbine under full load and glowing. The brightly glowing inlet branches will be noticed and the glowing turbine guide blades.

The photograph taken when illuminated shows the testing equipment: on the left, the nozzle which reproduces the wind during flight and on the right the reception funnel for the exhaust gases.

of the exhaust gases from modern petrol engines, a special combustion chamber and pipes to withstand temperatures up to 1100 ⁰ C were made.

This testing equipment allowed us to test charging sets at gas temperatures up to 1000 ⁰ C, at full speed, which, of course, heated up the turbine to bright red heat. Especially at night, the set when running presents an exceptional and fascinating appearance, the bright red guide blades being then clearly visible through the dark runner blades, which, at this speed, can no longer be distinguished.

The whole set is surrounded by the glowing gas piping and carries the mind back to some medieval machine.

(MS 774) *A. Meldahl. (Mo.)*

SELF-STARTING STEAM POWER STATIONS.

Decimal index 621.311.22.078

Stand-by and peak-load power stations must have short starting times. Velox steam generators and turbines designed for rapid starting fulfil this condition. However, the advantages of steam power stations of this kind can only be profited by fully when they are of the self-starting type. The description given here of a plant of this kind explains the simple design of the self-starting gear while records of service testify to the reliability of the plant.

Why are self-starting steam power stations desirable?

THE time necessary to start a steam power station from cold state plays an important part in the operation of stand-by and peak-load power stations, which are called on to participate in power delivery, both frequently and unexpectedly.

The time in question is very dependent on how soon it is safely possible to get pressure on the boiler. Limits to this are set chiefly by the brickwork, in the case of standard boiler design. It is frequently necessary to impose a starting time of more than an hour in the interests of the safety of the boiler plant. To this must be added the time necessary to put the turbo-set under load, which, however, is generally shorter; here a slow warming up at low speeds is often necessary in order to prevent inadmissible heat deformation of the turbine casing. The remaining starting operations are carried out, for the most part, simultaneously with those just specified. Thus, they hardly affect the total starting time and are performed manually by the station operators in proper sequence. As long as setting-to-work operations have to be slow, on account of the boiler and engines, the danger of skilled operators carrying out any wrong operations, imputable to human error, is not very great.

However, conditions change in Velox boiler plants. These boilers have no brickwork and work in conjunction with turbines specially designed for rapid starting. If full advantage is to be taken of them, it is necessary that the starting operations be carried out very quickly. The time available for this is insufficient with purely manual control despite the simplicity of the Velox starting process.

Engineers were, therefore, faced with the problem of extending the automatic operating qualities of the Velox power station so as to include the starting process as well, this in the case of those plants for which rapid taking over of load and service reliability are of paramount importance. Only thus was it possible to take full advantage of the properties of the Velox boiler in this respect. Short starting times are chiefly important in power stations which in their capacity of stand-by plants or peak-load plants are called on to take over load at irregular intervals and at undetermined times. In such plants, either the operators must be in constant readiness even when the station is shut down or else the time necessary for starting the plant will be lengthened considerably, in the case of unexpected demands, by the time necessary for the operators to reach their posts.

The automatic starting equipment, designed by Brown Boveri and built for the first time, in 1938, for a complete steam power house, is placed in a 10,000-kW station built for peak-load and emergency duty. The design is such that one single impulse suffices to actuate the starting process and to bring about the self starting of the auxiliary machinery in correct sequence. The starting impulse is imparted either by remote control from a control post situated at any distance away or else by failure of voltage on the line connected to the stand-by station, when it is a case of service disturbances of the transmission network. In both cases, and simultaneously with the starting impulse, the operators (in the case under consideration there is only one attendant) are informed by signal. The attendant is otherwise occupied somewhere near, or is in his home close by; he goes to the power station in which the starting process is proceeding quite independently of his presence.

The equipment of the power station.

The Velox boiler with an output of 50 t/h delivers steam to a steam-turbine set of 10,000-kW output. This power is stepped up from 6000 V at generator terminals to 20,000 V in a main transformer and is then led by high-voltage cable to the distributing station. The power necessary for station requirements is tapped from the high-voltage side of the transformer and led to station bus-bars after the voltage has been stepped down in a station transformer. This auxiliary transformer also serves to start up the power station when external power can be taken for this purpose from the network. For self-starting without using external power, there is a 290-kW Diesel-generator set available. If, during service, the voltage of the main generator should break down due to trouble on the line, the station transformer is cut off from the station bus-bars and a house turbo-set, which usually runs under no load, takes over auxiliary power delivery without any interruption. The house turbine is not coupled to an independent generator, but to the driving motor (designed as a synchronous motor) of the Ward-Leonard converter set. This set generates the direct current at variable voltage for the speed regulation of the charging set of the Velox boiler. Even under ordinary operating conditions, it may feed back power, coming from the gas turbine of the Velox, into the network, in which case the motor runs as a generator. If, as a result of trouble, the voltage on the station system fails, the speed drop of the house set which results causes the house turbine to be put to work and it then supplies through the motor, again running as a generator, all the auxiliary power required by the station. — Fig. 1 shows the connections of the station supply system with the chief connection points for the auxiliary motors and other power consumers.

When ready for service, the power station is at rest and cold. Every piece of apparatus which has to be brought to a given position in order

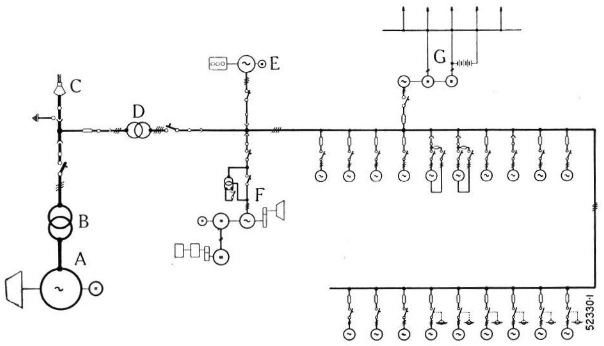


Fig. 1. — Station requirement plant of the automatic power station.

A. Turbo-generator 10,000 kW, 6000 V.
B. Main transformer 12,500 kVA, 6000/20,000 V.
C. Outgoing cable.
D. Station transformer 640 kVA.
E. Starting Diesel-generator set, 290 kW.
F. Ward-Leonard converter set with house turbine coupled to it.
G. Motor-converter set for charging storage battery.

to make the station ready has its own signal lamp mounted on a tell-tale (acknowledgement) board. After each shutting down of the station, the necessary preparatory measures to allow of starting it up automatically are taken. The main stop valves, the drain and sealing-gland steam valves of the turbine, all the valves of the condenser plant, the slide valves of the feed-water pump and pipe are opened, the condenser, condensate and feed-water pump are filled with water. All lamps on the tell-tale board are lighted up and thus show that the station is ready to start up. The

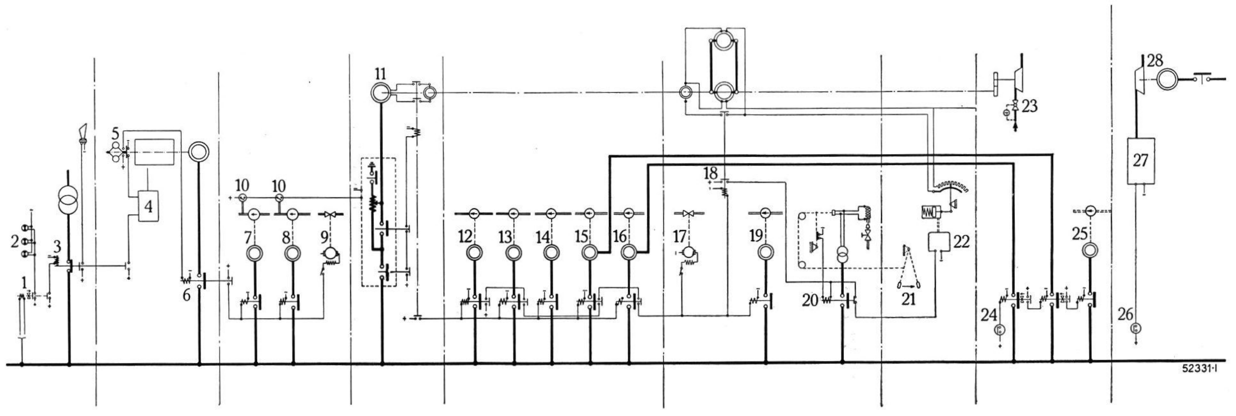


Fig. 2. — Fundamental diagram of connections for the automatic starting gear of a Velox steam power station.

1. Supervisory relay of the network voltage.
2. Emergency lighting.
3. Auxiliary transformer breaker.
4. Automatic starting device of Diesel engine.
5. Centrifugal switch on Diesel engine.
6. Breaker of Diesel generator.
7. Motor driving lubricating-oil pump of house turbine.
8. Motor driving lubricating and governing oil pump of the Velox boiler.
9. Motor of cooling-water valve.
10. Pressure relay.
11. Synchronous motor of Ward-Leonard converter set.
12. Motor driving auxiliary oil pump of main turbine.
13. Motor driving condensate pump.
14. Motor of high-pressure lubricating oil pump of main turbine (to facilitate turning over).
15. Motor driving feed pump.
16. Motor driving boiler-water circulating pump.
17. Motor of drainage valve of Velox boiler.
18. Field switch.
19. Motor driving fuel pump.
20. Contactor of ignition device of Velox boiler.
21. Ignition device of Velox boiler.
22. Running up device of Velox boiler.
23. Steam pressure relay of house turbine.
24. Steam pressure relay for switching over motors 15 and 16 to full speed.
25. Motor to drive cooling-water pump.
26. Steam-pressure relay for switching in the starting control of main turbine.
27. Running-up device of main turbine.
28. Main turbine.

attendant leaves the power house with the knowledge that if it is suddenly called on to deliver power, the sequence of operations for putting the plant to work will be initiated instantaneously and automatically.

Self-starting when the voltage on the system fails.

The diagram of Fig. 2 allows of following the principle of the self-starting process of the station. The network voltage supervisory relay 1 is the first to respond to a drop in the network voltage. This relay trips with a certain time lag, so that voltage fluctuations of short duration do not affect the station. After the time lag has elapsed, the emergency lighting 2 of the station, supplied from a storage battery, is switched in and the station-supply bus-bars are separated from the external network by breaker 3 of the auxiliary transformer. At the same time the hooter starts to warn the attendant. The automatic starting device 4 of the Diesel-engine is switched in and the Diesel engine started through the agency of compressed air. It attains full-rated speed in only a few seconds and is ready to take over load. If ignition should be unsuccessful at the first starting operation, the process repeats itself until the engine is really ignited.

When the engine has attained its rated speed, the centrifugal switch 5 connects the Diesel generator to the station supply bus-bars by the closing of switch 6. The auxiliary contact of this switch closes the contactors of motors 7 and 8 which drive the lubricating-oil pump of the house turbine and the lubricating and governing-oil pump of the Velox boiler. Motor 9, supplied directly by the battery current also starts up and opens the cooling-water valve of the elevated cooling-water tank from which the bearing and sealing-gland cooling is supplied until such time as the cooling-water pumps get running.

The pressure of the lubricating oil for the house turbine and Velox actuates both the pressure relays 10. As soon as these two oil pressures have built up, the contacts of the two relays close and bring about the starting of the synchronous motor of the Ward-Leonard converter set (which also serves as a house generator). The motor is started through a starting transformer in three steps. After attaining 80—90 % of synchronous speed, a centrifugal switch switches over to full voltage; thus, the motor first runs up to its rated speed as an induction motor after which the excitation is switched in. The house set is now running synchronously and, for the time being, unloaded with the auxiliary network. An auxiliary contact on the field switch effects the switching-in of a series of auxiliary motors, namely:—

Motor for the auxiliary oil pump of the main turbine 12
Motor for the condensate pump 13
Motor for the high-pressure lubricating pump of the main turbine, which facilitates turning over, 14.
Motor for the feed pump 15.
Motor for the boiler-water circulating pump 16.

The two last are pole-changing motors and are started up in the four-pole winding connection so as to run at half speed, by which means a saving is realized on the power demand during the starting process.

After the switching in of motors 12 to 16, motor 17 opens the emptying valve of the Velox boiler. The Velox which, for purposes of conservation, is kept quite full during the period it is being held in readiness for service, is now emptied down to the level "highest water level" on reaching which a contact on the water level indicator causes the changing-over of the valve motor connections and produces a stop to the emptying process of the Velox. At the same time, the field switch 18 switches in the excitation of the Ward-Leonard generator. The d. c. starting motor of the charging set of the Velox boiler begins to revolve. The governing-oil pressure of the Velox boiler which also controls the excitation of the Ward-Leonard machine is so adjusted by means of an oil escape device that the charging set driven by the starting motor runs at about 25 % of its rated speed. The same regulating impulse puts the fuel oil pump 19 to work. A contact on the field switch now switches in the transformer for the ignition device of the Velox boiler through contactor 20. The ignition device is composed of a ceramic resistance rod which is brought up to glow temperature by the passage of the current.

At this time, the fuel oil feed to the boiler is still closed so that no ignition can take place yet. However, all preparations for ignition have been made. The preceding programme of starting automaticity has brought about the proper water level in the boiler, has started the boiler water circulation and the compressor which delivers to the combustion chamber the necessary quantity of air for ignition purposes. The boiler feed pump is running. The fuel valve opens, ignition takes place. At the same time by means of the running up device 22 of the boiler both fuel supply and charging set are so regulated that no overloading of the Diesel engine, which is still carrying the whole load of the station requirements, can occur. A short time after ignition only, steam generation begins in the boiler. As soon as the boiler pressure has reached 3 kg/cm^2 gauge, the steam pressure relay 23 opens the nozzle valves of the house turbine. This now gives power to the station system which takes load off the Diesel engine. This allows of switching over the circulating water pump 16 and feed pump 15 to full speed which is carried out, at about 4 kg/cm², through the agency of steam pressure relay 24 and the switching in of the cooling water pump through the agency of the contactor of the motor 25.

All auxiliaries for the turbo-set are now running and the turbine can be started up. This takes place in function of the steam pressure in the boiler, starting beginning at a pressure of 5 to 6 kg/cm² gauge. The steam-pressure relay 26 switches in the automatic starting gear of the main turbine.

The running up proper of the turbine follows a "time-table"; the turbine is quickly brought up to a speed of about 1000 r. p. m. and allowed to run for a determined time at this speed to permit the rotor and housing to heat up, through and through. There is a time relay to prevent further acceleration and when it trips, the speed increases rapidly to the rated figure of 3000 r. p. m. The plant is now ready to take over load.

The attendant summoned by the warning signal emitted at the beginning of the starting period finds the station already in full operation when he appears. He can then switch the set on to the network which is in need of power and begin power delivery.

Experience gained in service.

The phases of the starting process take less time to perform than to describe. In order to get a true picture of how the automatic devices function, the processes during starting up were recorded by means of cinomatographic records of the supervisory instruments. Fig. 3 shows the time characteristics of the chief processes recorded in this manner experimentally. The following impressive time scale was thus shown:—

Time:

A.	0 min 00 s.	Plant in readiness for service, perfectly immobile and cold. A failure in network voltage causes the impulse to be given for the self starting of the plant.
B.	0 min 20 s.	The auxiliary Diesel set is running; it is already delivering power to the station-requirement system.
C.	1 min 25 s.	The boiler has been ignited, fuel consumption increasing.
D.	3 min 30 s.	Steam generation in boiler beginning.
E.	4 min 00 s.	Boiler under pressure of 6 kg/cm² gauge, turbine starts.
F.	5 min 30 s.	Boiler at rated pressure, steam temperature about 300° C.
G.	7 min 05 s.	Turbine at full speed.
H.	7 min 25 s.	Power station delivering power.
I.	10 min 00 s.	Power station delivering full output of 10,000 kW to network.

Thus, 10 minutes after the alarm signal was given, the Velox power station, which was unattended, at rest and cold, is delivering its full load of 10,000 kW.

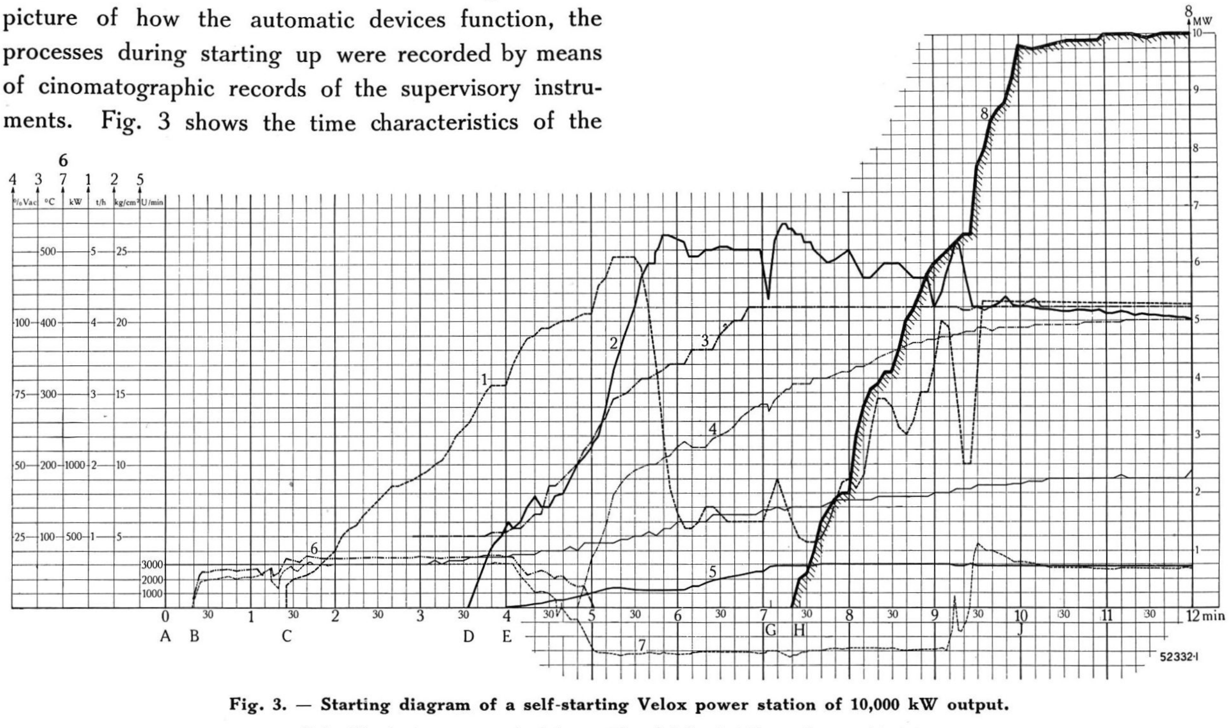


Fig. 3. — Starting diagram of a self-starting Velox power station of 10,000 kW output.

Only 10 minutes are required for getting full load delivery from cold state.

1. Amount of fuel t/h.
2. Live-steam pressure kg/cm² gauge.
3. Live-steam temperature.
4. Vacuum in condenser %.
5. Speed of turbo-set.
6. Output of starting Diesel engine kW.
7. Output of house generator set kW.
8. Output of main generator kW.

The results of carelessness, nervosity or of operators not being ready at their posts, all factors which must be looked for in a station which, perhaps, has to lie idle for long periods, are entirely eliminated.

welcome simplicity in operation as well. It has proved so reliable and useful that other duties can be confided to it besides emergency starting of the plant. Thus, it is utilized regularly and indeed always when it is

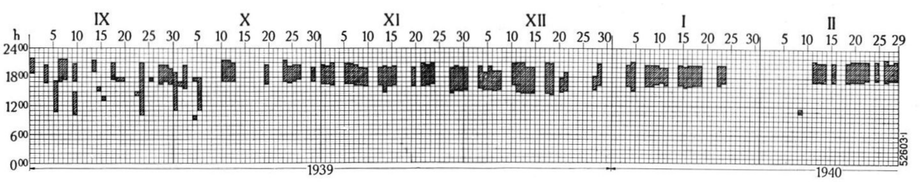


Fig. 4. — Section taken from the operating diagram of a stand-by and peak-load power station.
This plant was started up 98 times in the course of a half year.

Fig. 4 gives a section of a service diagram recorded on a typical peak-load plant equipped with automatic starting control gear. In the winter half of the year it is started once a day and often several times a day and is an emergency stand-by plant throughout the year. Self starting for a station of this type does not mean only rapidity in taking up duty, but a required that the station should cover peak loads. In this case the process is simply initiated by depression of a push-button. The starting control is then so set that the starting Diesel engine does not run, the power necessary for starting being tapped from the network through the station transformer.

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THE PRESENT-DAY DESIGN OF THE VELOX AS RESULT OF THE EXPERIENCE GAINED IN SEVERAL YEARS PRACTICE.

Decimal index 621.181.39

Ten years have passed since the fundamental tests were carried out which confirmed the Velox principle[1] and since the first Velox steam generator was built[1]. To-day, it can be justly claimed that the Velox principle proved sound from the very beginning and that its practical evolution did not meet with any great difficulties. If certain practical experience had first to be gained with the Velox steam generator, as indeed with every technical innovation, this practical experience did not go counter to the Velox principle itself, but revealed certain problems of detail, a phase through which all new boilers have to pass.

The purpose of this article is to describe the experience gained and what effect it had on modifying the definite design of the Velox.

THE slight modifications to which the Velox design has been subjected in the course of a decade chiefly affected the evaporator and superheater. The layout of these parts was made with a view to fulfil several requirements simultaneously, namely the lodging of the heating surfaces in the smallest possible space while retaining pipe bores common to ordinary boiler practice, elimination of brickwork and reduction of the number and dimensions of the housings which had to be made pressure proof in the Velox design. 3 to 7 fire tubes were grouped in a common water tube thus forming evaporator elements which, placed close to one another, line the inner wall of the combustion chamber (Fig. 1). Later on, the superheater was lodged in the combustion chamber itself in order to eliminate both the gas collector subjected to re-

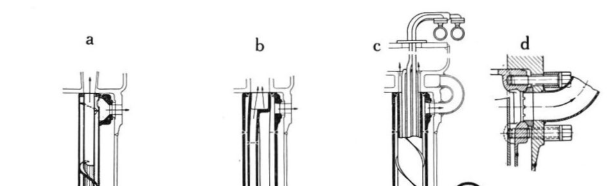


Fig. 1. — Evaporator element.

a. Single-tube element with inlet without a sealing gland. There is a common water-steam mixture outlet for two tubes.
b. Multi-tube element, three heating flues in a common tube.
c. Element with ring-shaped gas section with built in superheater.
d. Water-steam mixture outlet flange with lense-shaped seal, used with all three designs of the evaporator element.

The design a, utilized to-day, differs from the others by the gas flow, in a fire tube of ample diameter, meeting with no hindrances, further sealing glands are eliminated and the whole construction is simplified.

[1] It is assumed that the reader is acquainted with the Velox principle and the Velox steam generator. See one of the many articles on the subject, as, for example, that in this number, pages 189/191.

latively high temperatures and a separate superheater housing. To meet this object, built-in superheaters were developed and a superheater unit placed in each evaporator unit, the latter being generally lengthened so as to envelop the superheater unit. In order to have more evaporator surfaces and to attain the relatively small section of the gas tube requisite to a high velocity of the gas, another tube was built into the lower part of the evaporator unit which resulted in the hot gas duct now having a ring-shaped section. This compact design and the elimination of a separate housing for the superheater certainly represented a constructional advance, but from the point of view of operation it had certain disadvantages. The relatively narrow gas duct passage and sudden change in direction of flow to which the gas was subjected at the inlet to the evaporator unit caused choking of the gas passages when fuels were used which contained a considerable percentage of dust and ash. For use with fuels of this kind, the earlier design with separate superheater has been taken up again. To-day, the "single-tube" evaporator unit is given preference over the "multi-tube" evaporator unit. In the single-tube unit each evaporator tube has a single fire tube and 1 to 3 such pairs of tubes are combined to form an evaporation element having a common outlet for the mixture of steam and water. The stuffing boxes are eliminated. In earlier designs, these sealed off the water inlet pipe which protruded outwards from the interior of the combustion chamber; the water inlet pipes are now bolted to the lower water collector inside the combustion chamber (Fig. 2).

Liners.

In the first Velox units built, the combustion-gas collector, the duct to the superheater and the casing of the latter were provided with a liner of heat-proof metal sheeting generally carried on some cooling tubes or refractory brickwork. To-day, the liner consists of groups of tubes through which the circulating water flows and which form effective evaporation heating surfaces. As the hot gases are ejected at high velocity from the evaporator tubes and eddy round actively in the gas collector chamber and duct between it and the superheater, the coefficient of heat transmission (k) and, therefore, the heat transferred through the heating surface is very high, although the temperature of the gas has already dropped to something below 950 °C.

Turning the insulating liner of the superheater into an evaporation heating surface has the advantage that the quantity of heat converted is increased by 25 to 35 %. The temperature of the heating gases at the inlet of the superheater casing can, therefore, be made as much as 100° C higher, the gas temperature at the superheater outlet remaining the same. This rise in temperature allows of increasing the diameter of the fire tubes of the evaporator elements and reducing the heating surface of the superheater.

Originally, the only protection against temperature rise provided for the wall of the combustion chamber was formed by the evaporator elements which lined it. In later designs, a couple of millimeters of play were left between the tubes to facilitate dismantling them and the intermediate space behind the tubes was filled with refractory brickwork. In the latest Velox design a protective mantle of heat-proof metal sheeting is provided leaving a gap of 20 to 25 mm between it and the casing of the combustion chamber. This space forms a duct through which a part of the air to be used in the combustion process flows, exercising a cooling action.

Superheater.

As formerly, the superheaters are composed of several elements grouped together in bundles of about 6 to 10 pipe lengths (Fig. 3). The pipe lengths are slung in such a way that they are free to expand, being supported by sleeves and pins welded on the

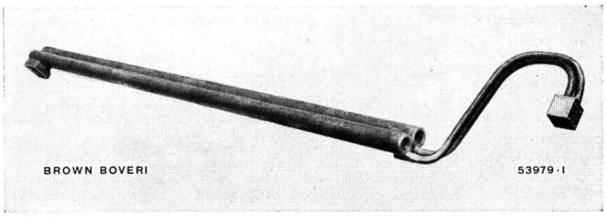


Fig. 2. — Double single-tube evaporator element.
With a common outlet for the mixture of water and steam.

tubes. The pipe connections, or bends, are formed either by pressing and welding the tube ends themselves, or by using cast-steel end pieces. From the point of view of flow properties, the cast-steel ends are better than pressed bends.

Economizer.

There are two economizer designs: the water-tube type in which the feed water flows through the tubes, the latter forming separate elements built into the economizer sheet-metal housing with a cover over its whole length; the gas-tube economizer in which the gases flow through the tubes and the tubes are welded into the end plates of the economizer.

Charging set.

Little change has been made in the charging set. Especial consideration was given to wear due to slag produced by the fuel. The flow conditions were improved within the turbine and attacks on the metal were countered by placing dust separators in front of the set or by designing the gas turbine casing itself so that it acted as a dust eliminator at the gas inlet end. The blades themselves are of heat-proof steel, usually V 2 AED steel specially hardened on the inlet edge.

Apart from improvements to the blading, resulting from investigation work and knowledge acquired in practice, which was chiefly expressed in a raising of the efficiency of these axial compressors, practically no changes were introduced to the compressor of Velox charging sets.

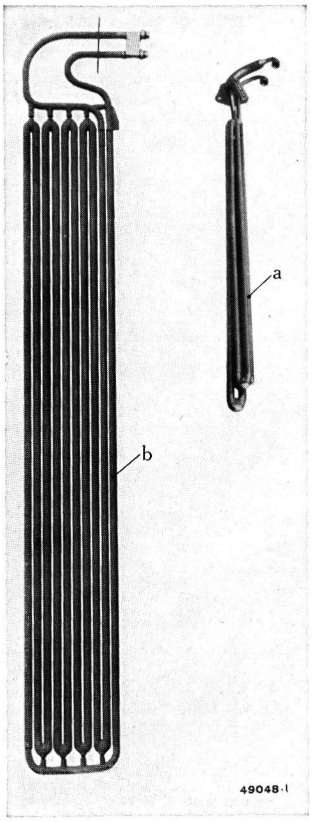


Fig. 3. — Superheater element.

a. Bundle of tubes for Velox with built-in superheater.

b. Flat bundle of tubes for Velox with a separate superheater.

An element b corresponds to about two elements a. The strongly-built tubes and their support by pins guided in sleeves will be noted.

Circulating pump.

As regards the circulating pump, the tendency of the shaft gland to leak was successfully remedied. It was chiefly due to the shaft running out of truth, owing to defective erection, unequal forces on the wheel periphery or different expansions of housing and bearings under the influence of the water which is, practically, at boiling point. The out of truth running produces unequal wear of the gland packing, to remedy which, the gland neck ring is usually made tighter and tighter. This, however, produces undue wear on packing and on shaft and makes the gland cooling less effective. To-day great care is devoted to accessibility, interchangeability and effective cooling of stuffing glands (Fig. 6).

In order to eliminate cavitation phenomena and, above all, to prevent the pump ceasing to deliver when the pressure in the boiler drops suddenly, the pump is connected directly to the separator by means of a branch designed like a diffusor; this allows a part of the speed of rotation of the water ring to be transformed into pressure. The speeds of inflow to the pump are kept low.

Water-steam separator.

It was found possible to reduce the diameter of the water-steam separator by building in certain devices by means of which the amount to be separated is divided up between the outer rotating water ring, which now extends over the whole height of the separator, and the inner water volume not in movement. In certain cases, the separator is equipped with a special steam dryer.

None of the structural changes made so far had to be effected on account of the firing under pressure, the big heat conversion, the separation of steam from water by centrifugal action or on account of the charging set. They were made on account of practical experience such as is gained with every type of steam generator and which is only too well known under the terms boiler scale, salt incrustation, corrosion and slag formation.

Feed water problems.

The Velox does not suffer from great sensitivity to boiler scale formation feared by some as a result of the high heat conversion coefficient nor is it, on the other hand, absolutely free from boiler scale formation as others expected on account of the rapid vaporization and high speed of water circulation; it behaves, despite its exceptional working conditions, very like any other high-duty boiler. It also requires as pure feed water as possible but is so designed to-day that it is more insensitive to transitory weaknesses in the feed-water supply than is the majority of high duty boilers.

Boiler scale formation over big parts of the heating surfaces have never been the cause of trouble. These deposits are revealed by the gradual rise in the gas temperature, which can be measured and kept under constant supervision at the inlet to the gas turbine. On the other hand, bulges in the tubes and cases of tubes bursting occurred on certain closely limited parts of the evaporator elements, due to insufficient circulation of water, to steam pockets and local scale formation. These defects have been satisfactorily eliminated by building in guide vanes which produce an effective flow of water against the tube walls.

Salt carried into the superheater is a more frequent occurrence than is boiler scale and it has given trouble to the designer of all modern high-duty boilers. In order that this should cause a defect in the tube, the deposit must be sufficient to heat the wall of the tube up to a localized glow temperature or else chemical reactions must set in which attack the wall, or else the amount of salt deposited is such that the circulation of steam is choked off.

By placing the superheater in a section of the heating-gas flow in which the lower tube ends, which are especially liable to salt deposits, are not subjected to higher temperatures than about 600 to 700° C, the tubes of heat-resisting steel are made practically immune to this danger. Thus, the Velox design with "separate" superheater is to be preferred to that with "built in" superheater in which the tube ends projecting down into the evaporator element are in a flow of heating gases which are still at a temperature of 1000^0 C.

Salt deposits which are harmless at moderate temperatures of the heating gases have nevertheless been the cause of trouble, as a result of insufficient experience. In order to remove the salt from the superheater it is subjected to a washing process, as is, indeed, well known. By sluicing the walls of the tubes the salt coating gets loosened and flows downwards where it collects in the tube ends in the form of a kind of thick pap. If the swilling out process is insufficient the salt incrustation does not dissolve and then the tubes get stopped up. The same thing can happen when the boiler is frequently put out of service at short intervals, the condenzation of the steam causes salt scale to loosen and drop down. Cloggings of this kind make themselves felt in the Velox by an increase in the pressure drop in the superheater and this allows of eliminating them before serious damage is done. Of course, the most radical means is to make sure that no salt gets into the superheater, in other words that the steam is perfectly dry.

The centrifugal water-steam separator delivers very dry steam in spite of its small size and the considerable volume of steam separated out. Numerous measurements carried out showed humidity of less than $^1/_2\,^0/_0$ in the steam. When the steam was wetter the cause was to be sought in insufficient attention having been given to the importance of sufficiently smooth separation surfaces. If, for example, the rapidly rotating water comes on roughness or some obstacle, it sprays upwards and particles are carried off by the steam already separated out. Wetness can also be caused by the water content foaming.

To what extent water reaches the superheater due to foaming depends very much on the water level, on the steam outlet and on the velocity of the water rotating in the separator. Just as the danger of trouble arising through salt being carried along depends on the design so does the carrying along of drops of water containing salt depend on constructive measures. This explains why different designs of Velox behaved very differently as regards the maximum allowable alcalinity of the circulating water and, for example, why Velox with a sodium number in excess of 1000 give no trouble while others cannot be operated at more than 500. The reason why sodium figures below 200 are advised even for Velox units with separate superheaters and high steam chamber in the separator, which would really allow of strong salt concentrations, is for the sake of conformity with common practice for modern high-capacity boilers.

In order to eliminate the last trace of wetness from the steam, special steam dryers have been evolved, as was said before. These can be built into the upper part of every Velox water-steam separator. They are made of sheet metal or are cast and form surfaces in the form of several screw thread channels. The steam flows at high speed inside the thread channels throwing off the drops of water as it goes, these creep down the surfaces and reach a drain pipe through which they flow off.

The reason why especial value is attached to the drying of the steam is not solely to protect the superheater but especially in order that the turbine blades should not get incrusted with salt.

In economizers, as is well known, the oxygen content of the water may lead to corrosion. High flow velocities of the water and smooth surfaces devoid of dead ends prevent oxygen collecting. In this respect the water-tube economizer is superior to the gas-tube type. Oxygen corrosion can be avoided in both types if the feed water, as is generally demanded to-day, is de-gassed. The oxygen content should be below 0·05 mg/l if possible. This is especially the case for boilers which are to be idle.

Chemical attacks can also take place in the economizer. Practice does not absolutely confirm former expectations that the tubes of a Velox are kept free of drops of condensate water from combustion thanks to the high velocity of flow. Therefore, there is no assurance that the Velox economizer will not corrode because the drops of condensate water from combustion and drops of sulphuric acid have no time to exert an aggressive action. However, it has been found, in practice, that the Velox economizer is indeed less subjected to corrosion of the gas side than is the case with steel economizers of ordinary boilers. When the Velox is under small partial loads, the velocity of the gases in the economizer is reduced. In order to make sure that the drops will be blown away in any case, it is advantageous that the heating gases should flow from top to bottom so that the force of gravity is added to the force of flow. This advantage can be especially utilized in the gas-tube economizers in which it has been found that the drops only collect at the lower tube edge and only exercise a corroding influence there. For this reason, the tubes are prolonged below the lower end plate, and these extra lengths of tube are allowed to corrode if any corrosive action manifests itself.

The fundamental Velox tradition that all heating surfaces should be interchangeable and replaceable by parts kept on stock can be applied to the water-tube economizer but not to the gas tube one which forms a single tubular body.

The most reliable protection against corrosion is, of course, the avoidance of too low temperatures on the gas and water sides. On the gas side modern practice goes down generally to about 160 ° C and on the water side to about 80 ° C. Both these figures could be reduced by 20 % easily, if especially high Velox boiler efficiency be called for. As regards the feed water, the inlet temperature is high in any case, when exhaust steam or dewatering steam is used or when a higher degree of preheating is demanded for cleaning the feed water and for deaeration.

The question of fuel.

As was mentioned at the beginning of this article, cloggings in the gas passages were the cause of the abandoning of evaporating units with built-in superheaters. In the early period of Velox developments (1932 until the end of 1935) no trouble of this kind was recorded. Later, this kind of trouble got more common as the fuel oil became of an increasingly poor quality, on account of the slag-forming impurities it held. As is known, crude oil can be soiled geologically by uncombustible bodies, it can also absorb salt on sea journeys. The most serious soiling, however, is that to which residue oil, which is almost exclusively used in the Velox, is subjected in the refinery by additions of soda and chalk etc. substances used, to-day, to refine crude oil and which remain in the residual oil. According to the quality of the impurities, the ash or slag which separates out after combustion of the oil in the form of tiny melted drops differs as regards fusion point, conglomeration, adherence and solubility in water.

The flow conditions of the heating gases which carry along the particles of slag are of essential importance as regards the trouble which the slag may cause. It has been shown that when the flow of the gases is deflected suddenly or if the flow strikes a partition or an obstacle the drops of melted slag are hurled outwards and stick to the cold surfaces gradually forming adhesive crusts or nests of slag. Obviously if the section of the hot gas flue is small it will get quickly choked. If, however, sharp deflections are avoided and high flow velocities only attained when the direction of flow is parallel to the smooth tube walls, the drops harden, roll along the walls of the flues and leave the smoke stack as small particles. The earlier evaporator elements with several tubes and non-deflected gas flow were never stopped up. The single-tube elements used to-day the fire-tube diameter of which is 50 to 95 mm, have still less tendency than ever to get choked up. In order to be able to use fuel containing much incombustible matter in the older type of plant with small heating-gas section as well, that is to be able to eliminate choking of the passages quickly, the combustion chambers were provided with washing de-

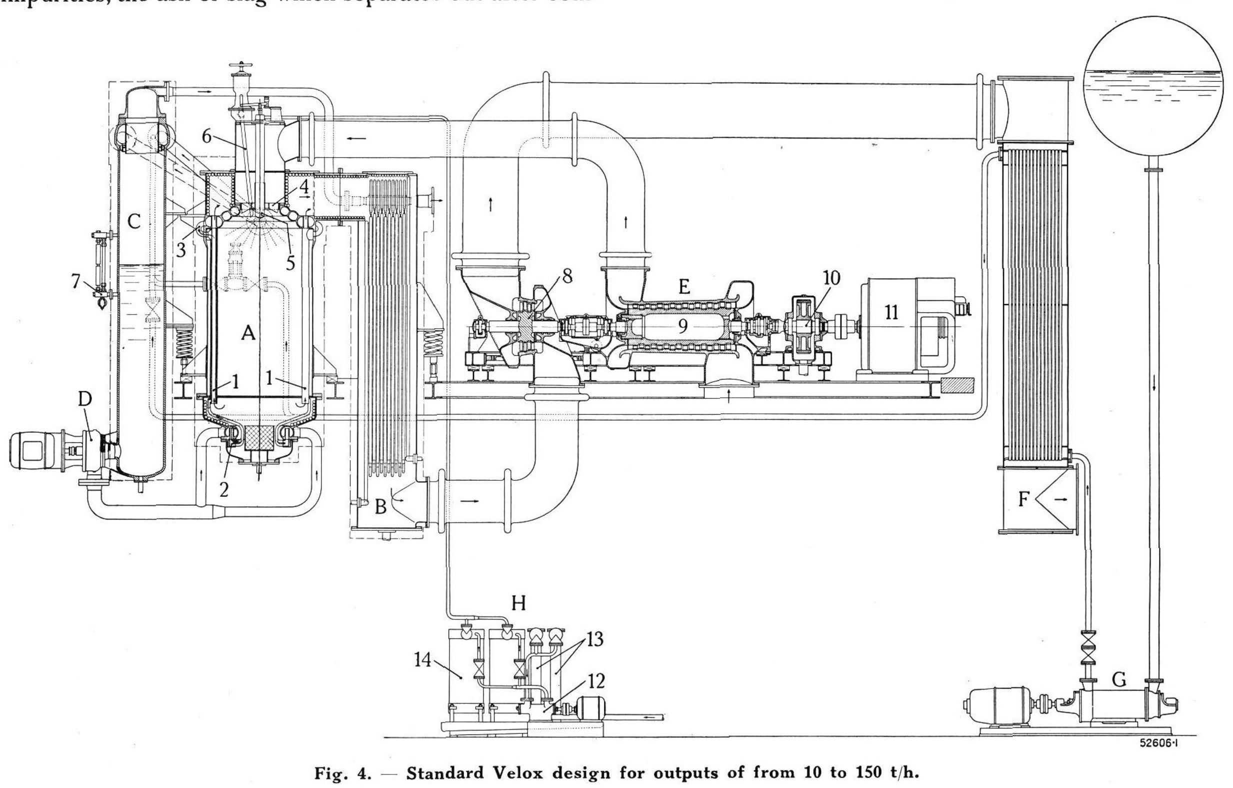


Fig. 4. — Standard Velox design for outputs of from 10 to 150 t/h.

A. Combustion chamber:—
1. Evaporator tube.
2. Inlet for circulating water.
3. Outlet for water-steam mixture.
4. Air nozzle ring (whirl vanes).
5. Burner.
6. Igniter.

B. Superheater.

C. Water-steam separator:—
7. Water gauge.

D. Circulating pump.

E. Charging set:—
8. Gas turbine.
9. Compressor.
10. Reduction gear.
11. Starting and regulating motor.

F. Economizer.

G. Feed-water pump.

H. Fuel supply:—
12. Fuel pump.
13. Filter.
14. Economizer.

vices. It must be remembered that the slag is composed in major part of salts which are soluble in water and are thus very amenable to elimination as the combustion chamber of the Velox can be filled with water. The slag then dissolves and can be drained off with the water after a couple of hours. As about two hours are necessary to cool down the boiler and only a couple of minutes to start it up again, the time during which service is interrupted is short, very different to the interruption of service which is necessary when slag chokes up "forced circulation" boilers and "once-through" boilers with closely spaced tube coils. Here cleaning is both difficult and long.

We have already indicated what is necessary to counteract the effect of ash and slag on the gas turbine.

The trouble on the heating gas side as on the water side can be considered as having been successfully eliminated. Further, poor qualities of feed water should no longer be considered as a cause of boiler defects. To-day means are available to make almost any water utilizable and to maintain it by phosphates or other means in sufficiently good condition.

General design.

Fig. 4 shows the modern Velox design. Apart from the scale, the illustration is valid for plants in a range of from 10 to 150 t/h. In plants for over 50 t/h, two superheater casings instead of one can be used. For boilers for less than 20 t/h, the single-tube evaporator elements can be ranged in a single row along the inner boiler wall. Above 20 t/h, two rows are necessary. According to the number of tubes the pitch varies, but the arrangement is always such that the tubes on the inner pitch circle always fill up the space between two tubes on the outer pitch circle. This arrangement and the perfect freedom to choose the most suitable length of superheater tubes allow of using tubes of the same diameters for boilers of very different outputs, which facilitates manufacturing, storing and standardizing spare parts.

Boilers to burn other fuels as, for example, poor gases (from blast-furnaces) are built to the same design as those for oil. The only difference is in the capacity of the combustion chamber and its burner; further blast-furnace gas Velox boilers have two blowers:— one for air and one for gas. Fig. 5 shows a combined burner for oil and high-grade gases.

Fig. 5. — Burner and air nozzle ring (whirl vanes) for oil and high-grade gases.

The oil is admitted through a central spraying nozzle and the combustion air enters between the whirl vanes, the gas inlet is through the hollow whirl vanes in the combustion chamber. Gas and oil can be turned on in turn as required during the operation of the Velox.

Excess power of the charging set.

It is always reckoned against the Velox that it does not permit of benifiting from a high feed water temperature obtained by feed heating by extraction steam, because the economizer cannot be replaced by an air preheater as the combustion air has already been heated by compression in the charging blower. Practically, this disadvantage is not of great importance because the efficiency of the Velox is already very high. Further the elimination of the higher steam extraction points make the plant much simpler. On the other hand, there is another possibility with the Velox of improving the overall efficiency and this by making the gas turbine of the charging set deliver power. In different Velox plants excess power has been obtained from the gas turbine under favourable circumstances as, for example, low intake air temperature, high compression efficiencies, etc. This excess power could be delivered to the supply system through the agency of the starting motor, which then ran as a generator, and through the Ward-Leonard set. Here the temperatures of the driving gases before the gas turbine were considerably below 500 °C. If, now, these temperatures are increased to values which are quite admissible to-day, of 560 °C for example, excess power can be made available under all conditions which suffices, at least, to cover the requirements proper of the entire steam plant. This supplementary output may amount to about 2·5 to 3 % of the plant output. If it be remembered that the generation of power for the drive of auxiliaries, through the main generator, calls for the expenditure of about 4000 kcal/kWh and that only 1150 to 1250 kcal are necessary per kWh of the gas turbine excess power, this supplementary output of 2·5 to 3 % means an improvement of the efficiency of the plant of 8 to 10 %. If the savings are also reckoned which are effected by the higher Velox efficiency, the considerably better economic characteristics of the Velox as compared to all other types of boiler is made very clear.

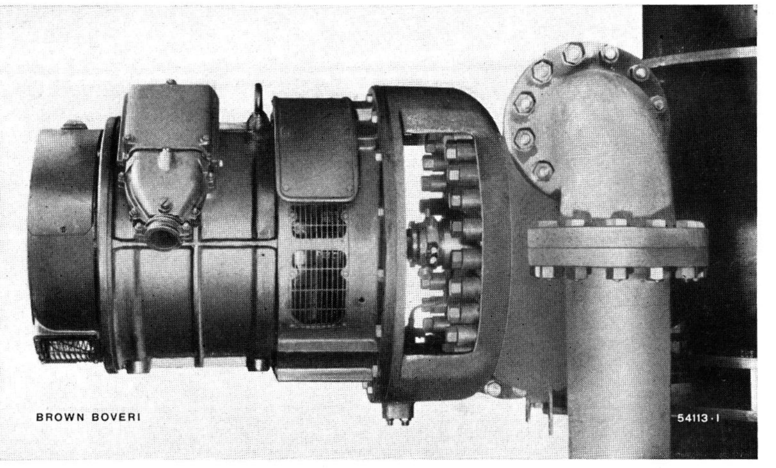
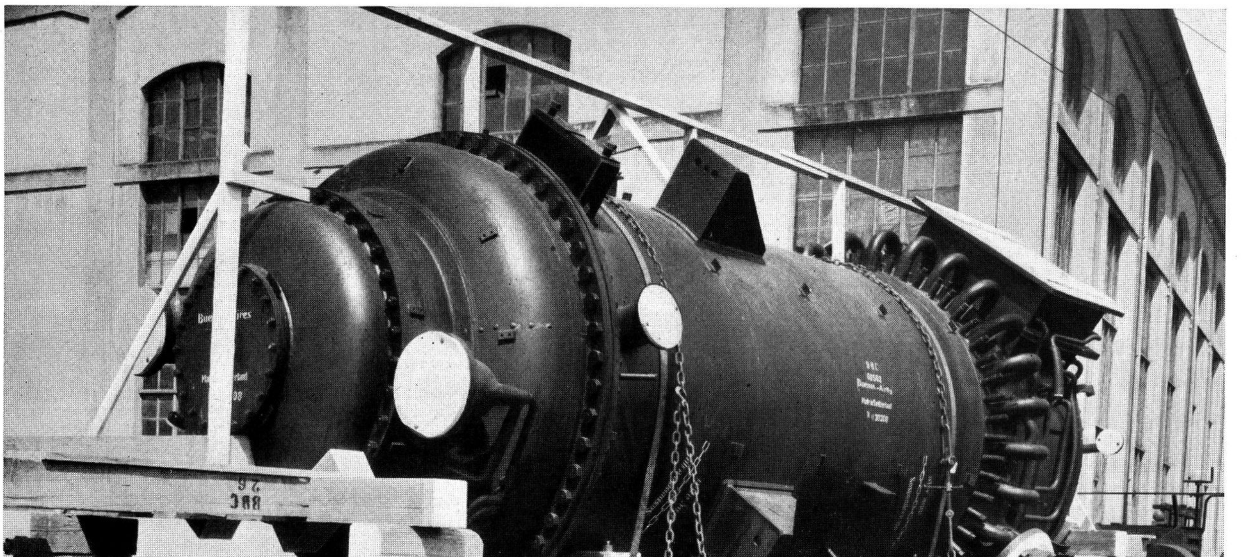


Fig. 6. — Circulating pump, built on to water-steam separator, with two symmetrical outlets.

Special designs.

Another special Velox design has been evolved for plants with very restricted space, such as on torpedo boats, destroyers and the like. It differs from that of the standard Velox in that not only the superheater but the major part of the evaporator heating surfaces are lodged in separate housings located horizontally below the boiler-room floor. The combustion chamber only contains the necessary wall lining and is made as small as is compatible with the maximum combustion chamber load. A first boiler of this new type will be running about the middle of this year. We intend to report thereon in due time.

(MS 766) *Dr. W. G. Noack. (Mo.)*



Transporting the combustion chamber of a 100 t/h Velox steam generator.

Velox boilers can be completely erected and tried out under full load in the maker's shops. They can also be dismantled and sent off on special trucks, in a few separate parts, even up to the biggest units (150 t/h of steam). Re-erection and putting to work on site, therefore, takes a short time only, for these reasons, and the erection work entailed is inexpensive.

THE DESIGN OF HEAT EXCHANGERS.

Decimal index 621.565.94.0012

A relation is first derived between the heat transferred and the energy loss in the case of turbulent flow, which enables a critical comparison to be made of the relative merits of different tube arrangements. In the second part, the conditions are determined, which decide the dimensions of, and the velocities in, the "most favourable" heat exchanger. The third part contains a reference to the correct economical dimensioning of heat exchangers.

In every case of heat transfer, there is a "most favourable" heat exchanger, which represents the best balance between the heat transferred, the energy loss, and the amount of material employed. In this study, the conditions are sought which such a "most favourable" heat exchanger must satisfy, the investigation being restricted to the case of turbulent flow.

1. *Heat transfer and friction work.*

Let us first seek a relation between the heat transferred and the energy loss. The problem is to determine the energy E which has to be expended in order that a quantity of heat Q may be transferred to a surface F, the average temperature difference being Θ.

Symbols used:

Q = heat quantity flowing per second,
G = weight of gas or fluid flowing per second,
α = heat transfer coefficient,
c_p = specific heat at constant pressure,
Δt = temperature variation of medium during flow,
w = velocity of flow,
ζ = coefficient in pressure drop formula,
l = tube length,
d = tube diameter,
γ = specific weight,
μ = absolute viscosity,
λ = thermal conductivity

System of units: m, kg, sec.

The heat given up to the surface F is:

$$Q = \alpha \cdot \Theta \cdot F, \tag{1}$$

and the gas loses a corresponding amount of heat

$$Q = G \cdot c_p \cdot \Delta t, \tag{2}$$

whence,

$$\frac{\Delta t}{\Theta} = \frac{\alpha F}{G c_p} \tag{3}$$

It is convenient to derive the expression for the energy loss first for longitudinal flow through the tubes, as the result can later readily be applied to the case of cross flow over a tube bank. The expression for the pressure drop in a tube is

$$\Delta p = \zeta \frac{l}{d} \cdot \frac{w^2}{2g} \gamma \tag{4}$$

Putting the free gas section equal to f, and remembering that

$G = w \gamma f$, and further that $\frac{l}{d} = \frac{F}{4f}$, we obtain from equation (3)

$$\frac{\Delta p}{\Delta t/\Theta} = \zeta \frac{w^3 \gamma^2 c_p}{8 g \alpha} \tag{5}$$

The ratio of the pressure drop to the heat transfer depends, therefore, to a large extent on the velocity with which the heating surfaces are swept. If we group together those quantities which depend on the velocity, and introduce the Reynolds number

$Re = \frac{w \gamma d}{\mu g}$ and the Nusselt number $Nu = \frac{\alpha d}{\lambda}$, we obtain the following relation between the heat transfer and the pressure drop:

$$\frac{\Delta p}{\Delta t/\Theta} = \frac{\mu^3 g^2 c_p}{8 \gamma \lambda d^2} \cdot \zeta \cdot \frac{Re^3}{Nu}.$$

Putting also $\zeta \frac{Re^3}{Nu} = Z$, (6)

enables us to write the desired relation between the heat transfer and the energy loss in the case of longitudinal flow through a tube, referred to 1 kg of medium flowing through the exchanger as

$$\frac{E}{\Delta t/\Theta} = \frac{\mu^3 g^2 c_p}{8 \gamma^2 \lambda d^2} \cdot Z \tag{7}$$

Similar expressions may be obtained for the case of a tube bank with cross flow, if ζ is taken as denoting the pressure drop coefficient per tube row. If the bank is z_1 rows deep, the expression for the pressure drop becomes

$$\Delta p = \zeta \cdot z_1 \cdot \frac{w^2}{2g} \gamma, \tag{4a}$$

where w is the velocity at the narrowest point between the tubes. Further, if $s \cdot d$ denotes the pitch of the tubes across the flow, and z_q denotes the number of tubes per row also in the direction across the flow, then with $F = \pi\, d\, l\, z_q \cdot z_l$ we obtain

$$\frac{\Delta p}{\Delta t/\Theta} = \frac{w^3 \gamma^2 c_p}{8 g \alpha} \cdot (s-1) \cdot \frac{4}{\pi} \zeta, \tag{5a}$$

and introducing Re and Nu gives

$$\frac{\Delta p}{\Delta t/\Theta} = \frac{\mu^3 g^2 c_p}{8 \gamma \lambda d^2} \cdot (s-1) \cdot \frac{4}{\pi} \cdot \zeta \cdot \frac{Re^3}{Nu}$$

Let us put

$$(s-1)\,\frac{4}{\pi}\cdot\zeta\cdot\frac{Re^3}{Nu}=Z_q \qquad (6a)$$

and obtain thus for the case of cross flow the equation corresponding to equation (7)

$$\frac{E}{\Delta t/\Theta}=\frac{\mu^3\,g^2\,c_p}{8\,\gamma^2\,\lambda\,d^2}\cdot Z_q \qquad (7a)$$

giving the relation between the energy loss and the heat transfer for cross flow.

It is seen that the first term of the equations (7) and (7a) contains characteristic quantities of the medium and the tube diameter. This means that for a given tube diameter the heat exchanger is fully characterized by the number Z. Now the pressure drop coefficient ζ is function of Re, whilst Nu is a function of Re and of the Prandtl number, if we neglect the transition zone at the inlet, which is entirely permissible with cross flow exchangers many rows deep, or with longitudinal flow exchangers with relatively long tubes. But since the Prandtl number is purely a function of characteristic quantities of the medium, and we are interested only in a comparison of heat exchangers working with the same medium, and operating within the same temperature limits, Nu will depend only on Re. This makes it possible to plot Nu as a function of Z. The Nu — Z diagram,

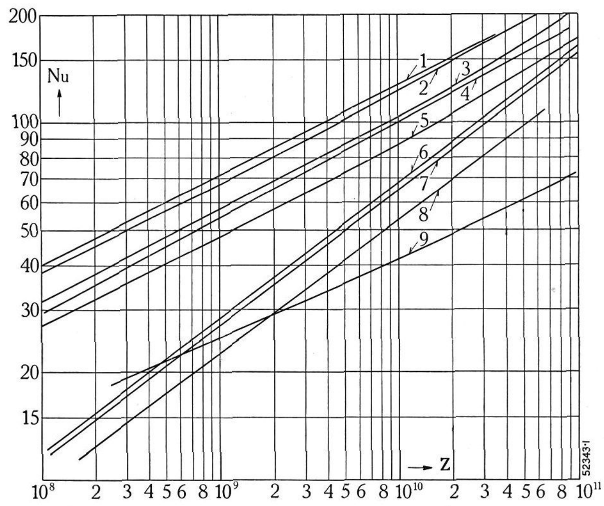


Fig. 1. — Relation between the energy loss and the heat transfer.

1. Crossflow, staggered tubes, pitch 1·25 × 1·25.
2. Crossflow, tubes in line, pitch 1·25 × 1·25.
3. Crossflow, tubes in line, pitch 1·5 × 1·5.
4. Crossflow, tubes in line, pitch 2 × 2.
5. Crossflow, tubes in line, pitch 3 × 3.
6. Longitudinal flow in a tube.
7. Longitudinal flow between tubes, pitch 1·5 × 1·5.
8. Longitudinal flow between tubes, pitch 2 × 2.
9. Single tube in crossflow.

The pitch is expressed as times the tubes diameter.

The diagram shows, for some common tube arrangements, the relation between the heat transfer number Nu and characteristic number Z for the energy loss. For any given arrangement of the tubes there corresponds to every value of the heat transfer number Nu a definite value of Z, with the help of which the pressure drop and the exchanger surface may be obtained from the equation (7) or (7a).

therefore, gives a clear picture of the merit of a heat exchanger surface. Tube banks of different pitch are represented by different curves in the Nu — Z diagram. The higher a curve lies, the greater may be the heat transfer loading for a given energy expenditure, or, conversely, for a given surface and heat loading, the smaller the energy expenditure. Fig. 1 which is drawn for gases, contains curves relating to some of the most frequently used tube arrangements. In the case of cross flow, they are based on the values derived by Grimison[1] from the tests of Pierson and Huge. To enable a comparison to be made, curves for longitudinal flow have been inserted calculated with the aid of the formula given by Jung[2]. It is seen that only at very high rates of heat transfer such as those which are achieved by the very high gas velocities attained in the Velox boiler, does the longitudinal arrangement become more advantageous than the cross flow one. The curves giving the values of Z which in Fig. 1 are plotted to a logarithmic scale are very nearly straight. It is, therefore, permissible when considering segments of these curves, and without introducing any appreciable error to assume the following relation:

$$\log Z = B + m \log Nu$$

$$\text{or,}\quad Z = B\ Nu^m, \qquad (8)$$

where B and m are constants whose value depends on the position in the diagram of the segment under consideration.

Equation (7) can, therefore, in the case of the segment concerned and with the aid of equation (3) be transformed and the expression for the power loss written

$$L=\frac{\mu^3\,g^2}{8\,\gamma^2\,d^3}\cdot F\cdot B\cdot Nu^{m+1} \qquad (9)$$

8. *The condition for the most favourable exchanger.*

The merit of an exchanger can only be judged when it is known what quantity of heat is equivalent to the mechanical energy which has to be supplied in the form of compressor or pump work to overcome the resistance of the exchanger. If the exchanger is an air preheater forming part of a steam power unit or of a gas turbine then the overall efficiency of the plant or that which the plant is estimated to have with the exchanger in service, determines the ration of the energy (expressed in kcal), to the heat consumption required to produce this energy.

On the other hand, if it is a case of plain heat transfer for instance, in blast heaters, furnaces, etc., where the

[1] Correlation and Utilization of New Data on the Flow Resistance and Heat Transfer for Cross Flow of Gases over Tube Banks by E. D. Grimison, Trans. ASME. Oct. 1939, page 583.

[2] Jung, VDI Forschungsheft 380.

energy absorbed in overcoming the resistance of the exchanger has to be supplied in the form of power purchased from an outside supply, then the cost of this power must be balanced against the production cost of the more or less completely transferred heat in the exchanger. If η denotes the efficiency of the plant by means of which the heat energy of the fuel is converted into mechanical energy, then the economic performance of the exchanger is given by:

$$\text{Useful heat} = Q - \frac{AL}{\eta}$$

The useful heat is, therefore, equal to the difference of the heat transferred and the heat required for the production of the mechanical work absorbed.

The "most favourable" heat exchanger is, therefore, the exchanger which with a given surface and with a given diameter of tubes will result in a maximum amount of useful heat. The condition for this is

$$\eta\, dQ - A dL = 0 \tag{10}$$

We shall now seek expressions for dQ and dL in terms of Nu.

Let the suffix 1 denote the hot medium, and the suffix 2 the cold one. The meaning of the symbols is made clear by the Fig. 2. We may write for the temperature variation of the hot medium

$$t_1' - t_1'' = \varepsilon_1 \cdot (t_1' - t_2'),$$

and similarly, that of the colder one is

$$t_2'' - t_2' = \varepsilon_2 \cdot (t_1' - t_2').$$

Further, let the mean temperature difference be given by

$$\Theta = a \cdot (t_1' - t_2')$$

The factor a depends only on ε_1, and ε_2, and is plotted in Fig. 3 for counter flow, for cross flow and parallel flow, for the case $\varepsilon_1 = \varepsilon_2$. If we denote by k the overall heat transfer coefficient, the heat transferred is given by

$$Q = a \cdot k \cdot F\, (t_1' - t_2') \tag{11}$$

For a small change in the rate of heat transfer we

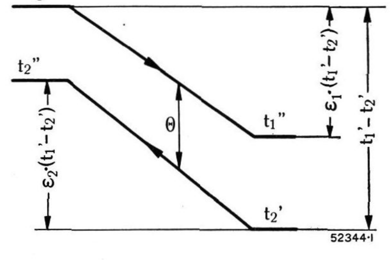


Fig. 2. — Diagram of the temperature variations in an exchanger.

have, since k F may be treated as a single quantity

$$dQ = \left(a + kF\frac{da}{dkF}\right) \cdot dkF \cdot (t_1' - t_2')$$

We put

$$dQ = b \cdot dkF \cdot (t_1' - t_2') \tag{12}$$

where b depends only on ε_1 and ε_2. From equations (11) and (12), and from the relation $Q = \varepsilon G\, c_p\, (t_1' - t_2')$, it is readily found that

$$b = \frac{a}{1 - \frac{\varepsilon}{a}\frac{da}{d\varepsilon}} \tag{13}$$

it being immaterial whether ε_1 or ε_2 is inserted. Curves for b are given in Fig. 3. In the case of heat transfer

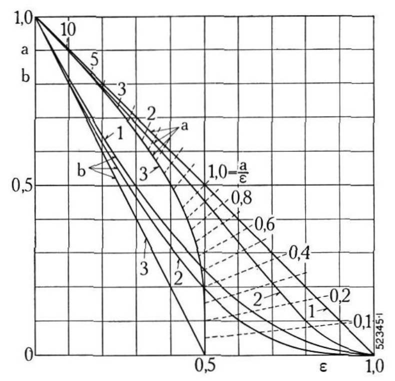


Fig. 3. — Characteristic numbers a and b of equations 11 and 12 for $\varepsilon_1 = \varepsilon_2$ in the case of counterflow, crossflow and parallelflow.
1. Counterflow. 2. Crossflow. 3. Parallelflow.

in metal exchangers the thermal resistance of the exchanger wall may be neglected without introducing any appreciable error. Hence, it is permissible to write

$$\frac{1}{kF} = \frac{1}{\alpha_1 F_1} + \frac{1}{\alpha_2 F_2} \tag{14}$$

Differentiating and introducing the Nusselt number in place of the quantities $d\alpha_1$ and $d\alpha_2$ gives

$$dkF = \left(\frac{kF}{\alpha_1 F_1}\right)^2 \cdot F_1 \cdot \frac{\lambda_1}{d_1} \cdot dNu_1 + \left(\frac{kF}{\alpha_2 F_2}\right)^2 \cdot F_2 \cdot \frac{\lambda_2}{d_2} dNu_2 \tag{15}$$

The total power loss is the sum of the losses for the hot and cold mediums. We use equation (9) and put

$$F\, \frac{\mu^3 \cdot g^2 \cdot B}{8\, \gamma^2\, d^3} = P \tag{16}$$

and obtain for the total energy loss

$$L = P_1 \cdot Nu_1^{m_1+1} + P_2 \cdot Nu_2^{m_2+1}$$

The change in loss with change of velocity is then given by

$$d\,L = P_1\,(m_1+1)\,Nu_1^{m_1}\,d\,Nu_1 + P_2\,(m_2+1)\,Nu_2^{m_2}\,d\,Nu_2 \quad (17)$$

The change in heat quantity transferred with change of specific loading kF is given by equation (12). Inserting equations (12) and (17) in equation (10) and taking into account equation (15) gives

$$\frac{\eta\,b}{A}\,(t_1'-t_2')\left[\left(\frac{k\,F}{\alpha_1\,F_1}\right)^2\cdot F_1\cdot\frac{\lambda_1}{d_1}\cdot d\,Nu_1 + \left(\frac{k\,F}{\alpha_2\,F_2}\right)^2\cdot F_2\cdot\frac{\lambda_2}{d_2}\,d\cdot Nu_2\right] - P_1\,(m_1+1)\,Nu_1^{m_1}\,d\,Nu_1 - P_2\,(m_2+1)\,Nu_2^{m_2}\,d\,Nu_2 = 0 \quad (18)$$

The coefficients of $d\,Nu_1$ and $d\,Nu_2$ must each equal 0. This gives two new equations namely,

$$\frac{\eta\,b}{A}\,(t_1'-t_2')\left[\left(\frac{k\,F}{\alpha_1\,F_1}\right)^2\cdot F_1\cdot\frac{\lambda_1}{d_1}\right] = P_1\,(m_1+1)\cdot Nu_1^{m_1} \quad (19a)$$

$$\frac{\eta\,b}{A}\,(t_1'-t_2')\left[\left(\frac{k\,F}{\alpha_2\,F_2}\right)^2\cdot F_2\cdot\frac{\lambda_2}{d_2}\right] = P_2\,(m_2+1)\cdot Nu_2^{m_2} \quad (19b)$$

Dividing equation (19b) by equation (19a), inserting for P_1 and P_2 the values given by equation (16) and substituting Nu for α_1 and α_2 gives

$$\frac{Nu_2^{m_2+2}}{Nu_1^{m_1+2}} = \frac{(m_1+1)\cdot B_1\cdot F_1^2\,\mu_1^3\,\gamma_2^2\,\lambda_1\,d_2^4}{(m_2+1)\cdot B_2\cdot F_2^2\,\mu_2^3\,\gamma_1^2\,\lambda_2\,d_1^4} \quad (20)$$

This equation determines the ratio of the velocities in the most favourable heat exchanger; it does not, however, say anything about the absolute value of the velocities. It means that heat exchangers in which this ratio of the velocities is observed have for a given surface and a given heat quantity the lowest friction loss.

If we take the roots of equations (19a) and (19b) and adding, and remembering that $\frac{k\,F}{\alpha_1\,F_1} + \frac{k\,F}{\alpha_2\,F_2} = 1$, we obtain as the second condition for the most favourable heat exchanger:

$$\sqrt{\frac{\eta}{A}\,b\,(t_1'-t_2')} = \sqrt{R_1\,Nu_1^{m_1}} + \sqrt{R_2\,Nu_2^{m_2}} \quad (21)$$

The factor R contains only constants.

$$R = (m+1)\cdot\frac{B\cdot\mu^3\,g^2}{8\,\gamma^2\,d^2\,\lambda}$$

The two equations (20) and (21) completely determine the most favourable heat exchanger. The resulting transcendental equation must be solved by trial. The surfaces are found with the aid of the numbers Nu_1 and Nu_2 and since Nu is a function of the Reynolds number, the velocities w_1 and w_2 are also determinate. The dimensions of the heat exchanger are, therefore, fixed.

3. *The correct economic dimensioning of a heat exchanger.*

It was seen in the first part that there is a function Z which serves as a criterion of the merit of tube arrangements in heat exchangers. In the second part the conditions are found which fix the dimensions of the surface and sections of the most favourable heat exchanger. The exchanger should, however, like every other apparatus be correctly dimensioned from the economic point of view, that is to say, the sum total of the capital charges and of the running costs should be a minimum.

If P denotes the capital cost, n the interest and depreciation rate, then the capital charges are

$$K_1 = n\,P,$$

and if L is the power absorbed, η the efficiency, h the yearly operating hours and p the price per kilowatt-hour, then the power costs are

$$K_2 = L\,h\,\frac{1}{\eta}\,p,$$

and the total yearly costs

$$K_1 + K_2 = n\,P + h\,\frac{1}{\eta}\,p\,L,$$

these should be a minimum hence,

$$n\,dP + h\,\frac{1}{\eta}\,p\,dL = 0 \quad (22)$$

The capital costs will increase approximately in proportion to the exchanger surface, and hence, for a given tube diameter, roughly inversely proportionally to the heat transfer number, or

$$F = Nu^{-1} \text{ and } P \sim Nu^{-1}$$

But according to equation (9), the energy loss is proportional to $F\,Nu^{m+1}$; hence substituting for $F = Nu^{-1}$

$$L = C\,Nu^m \text{ or } P \sim L^{-\frac{1}{m}}$$

differentiating and dividing by P

$$\frac{dP}{P} + \frac{1}{m}\,\frac{dL}{L} = 0 \quad (23)$$

Dividing equation (22) by equation (23)

$$n\,P + h\,\frac{1}{\eta}\,p\,m\,L = 0 \quad (24)$$

That is to say the total yearly costs are a minimum when the capital charges amount to m-times the power costs. Within the range of practical application, that is, for Nu = 40 to 120 the curve which over this range can be looked upon as a straight line, for instance curve 3 (cross flow heat exchanger with a tube pitch $1 \cdot 5 \times 1 \cdot 5$) gives an exponent $m = 3 \cdot 84$ and a constant $B = 166$; for curve 7 (longitudinal flow with a tube pitch $1 \cdot 5$ d) the figures are $m = 2.67$ and $B = 99\,500$. Accordingly the capital charges should amount in the case of a cross flow exchanger to 4 times and in a longitudinal flow one to about 3 times the power costs.

The starting point for this study has been the assumption of a fixed tube diameter and tube pitch. These and the choice between *staggered* or *straight* arrangement of the tubes are determined by dirt deposit and cleaning considerations. How closely these assumptions and the results of the calculation of the "most favourable" heat exchanger may be adhered to in practice depends on manufacturing conditions, but in any case the above exposition will serve as a guide to show in what direction and to what extent modications are desirable.

(MS 767) *K. Niehus. (Hv.)*

THE DETERMINATION OF THE VARIATION OF STATE IN TURBO-MACHINERY BY MEANS OF THE INCREASE IN ENTROPY.[1]

Decimal index 536.7 : 621—135

Thermodynamic relations are derived connecting the increase in entropy in turbo-machinery with the efficiency, which enable the change in condition to be determined more easily and more rapidly than is possible by the methods hitherto employed. In the case of gases in particular, the new method enables an entropy diagram to be constructed in which for a constant stage efficiency, the condition curve appears as a straight line.

1. *Introduction.*

Difficulties are encountered in the design of multi-stage turbines and compressors, due to the peculiar characteristics of real gases and vapours. Only in the case of a hypothetical perfect gas with a constant specific heat c_p, is it possible to determine the change in state by purely analytical means. Already with a semi-perfect gas, notwithstanding the fact that it obeys the relation $pv = RT$, the specific heat is a function of the temperature, and the assumption that it is constant is justified only if the pressure ratio is small.

In the following, a relation is, therefore, derived by means of which the end point of a polytropic expansion curve may, even in the case of vapours, be determined by a simple process. For perfect gases, the method yields mathematically exact results, for vapours, a close degree of approximation is attained. In all cases, the change in condition is determined with an accuracy sufficient for all practical purposes, and so quickly as to make its general adoption very desirable.

2. *Definition of the polytrope.*

It is necessary to explain first what is to be understood by the expressions polytrope and polytropic efficiency as used in the present article, because a polytropic variation of state is not always defined in the same way by different authors.

The polytropic efficiencies η_E and η_K, for expansion and compression respectively, are here used to denote the ultimate stage efficiency of a thermally insulated machine when the number of stages becomes infinitely large, i. e. the efficiency of a thermally insulated stage working between infinitely close pressure limits. Thus in the case of expansion

$$\eta_E = \frac{-\,di}{-\,A\,v\,dp}$$

and in the case of compression

$$\eta_K = \frac{A\,v\,dp}{di}.$$

In practice, the polytropic efficiency does not differ to any appreciable extent from the efficiency of a real stage η_{st} of finite but small pressure difference, and the turbine designer for instance, may take $\eta_{st} \approx \eta_E$ without introducing any error.

By polytrope, we understand a variation of state during which the ratio of di to Av dp remains constant, or

$$\frac{di}{Av\,dp} = \eta_E = \frac{1}{\eta_K} = \text{const.} \qquad (1)$$

The variation of state inside a multi-stage thermally insulated turbo-machine is, therefore, by definition polytropic if all the stages operate with the same efficiency.

This definition agrees with that given by Stodola[2], but not with that given by Schüle[3], who takes the ratio of dq to dT as constant. The difference between the two definitions is particularly noticeable in the case of vapours where according to Schüle's definition in addition to the adiabatic, any isothermal becomes a polytrope, whereas according to Stodola, adiabatics, isobars (compressor with $\eta = 0$) and throttling curves (turbine with $\eta = 0$) are looked upon as polytropes.

[1] A list of the symbols used with their meanings is given at the end of the article.

[2] A. Stodola, Dampf- und Gasturbinen, 5th edition, page 41.

[3] W. Schüle, Techn. Thermodynamik, 4th edition, Vol. I, page 118.

3. *The entropy increase of the polytrope.*

From the first and second laws of thermodynamics we have

$$dq = di - Av\,dp,$$

and

$$ds = \frac{di}{T} - A\frac{v}{T}\,dp. \qquad (2)$$

Inserting the value of di given by equation (1) yields

$$ds = \frac{1}{T}\cdot\eta_E\cdot Av\,dp - A\frac{v}{T}dp = -(1-\eta_E)A\frac{v}{T}dp \quad (3)$$

This relation says simply that the increase in entropy ds is equal to the loss of available work $-(1-\eta_E)$ Av dp, divided by the temperature T.

Further, it is readily seen from equation (2) that for i = constant,

$$-A\frac{v}{T}\,dp = \left(\frac{\partial s}{\partial p}\right)_i\cdot dp, \qquad (4)$$

and hence equation (3) may be written

$$ds = -(1-\eta_E)\cdot A\frac{v}{T}\,dp = (1-\eta_E)\cdot\left(\frac{\partial s}{\partial p}\right)_i dp \qquad (3a)$$

So far the treatment has involved applications of the fundamental laws only, so that expression (3a) gives a generally valid relation connecting the increase in entropy and the polytropic efficiency. It applies, therefore, to both gases and vapours.

4. *The polytrope for gases.*

For gases the quantity $\frac{v}{T}$, and hence also $\left(\frac{\partial s}{\partial p}\right)_i$ depends only on the pressure and it can, therefore, be directly integrated between the limits of any initial pressure p_1 and final pressure p_2

$$\Delta s = (1-\eta_E)\cdot AR\log\frac{p_1}{p_2} \qquad (5a)$$

$$= (1-\eta_E)\cdot(\Delta s)_i, \text{ for expansion,}$$

$$\Delta s = \frac{1-\eta_K}{\eta_K}\cdot AR\log_e\frac{p_2}{p_1} \qquad (5b)$$

$$= -\frac{1-\eta_K}{\eta_K}\cdot(\Delta s)_i \text{ for compression.}$$

In the above, $(\Delta s)_i = AR\log_e\frac{p_1}{p_2}$ is the difference in entropy between the isobars p_1 and p_2 at constant enthalpy i. It should be noted that in the case of gases, this relation is strictly true, that is to say, it involves no assumption as to the dependence of the specific heat on the temperature.

Figs. 1 and 2 show how to find in the T-s diagram the end point of a polytrope on any particular isobar, or conversely, how the polytropic efficiency may be calculated from the test data of any turbo-machine.

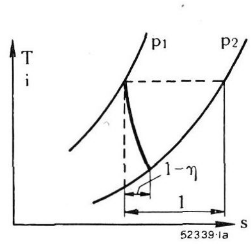


Fig. 1. — Expansion polytrope in the entropy diagram for gases.

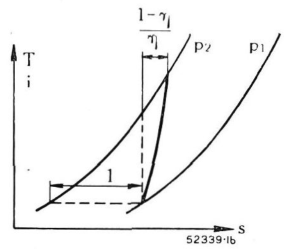


Fig. 2. — Compression polytrope in the entropy diagram for gases.

The diagrams show the simple relationship between the polytropic efficiency and the real increase in entropy expressed as a fraction of the entropy difference along the line i = constant.

5. *The polytrope in the i-s diagram for vapours.*

In the case of gases which are not perfect, i. e. in the case of vapours, equation (3a) cannot be integrated directly. In practice, however the relations (5a) and (5b) can still be used if $(\Delta s)_i$ is replaced by $\overline{(\Delta s)_i}$ which is thereby defined as the mean entropy difference of the isobars p_1 and p_2 at constant enthalpy i. This $\overline{(\Delta s)_i}$ is measured approximately in the middle of the condition curve in the manner shown by the example illustrated in Fig. 3. To be strictly accurate, all the elementary lengths denoted by $\left(\frac{\partial s}{\partial p}\right)_i\Delta p$ along the condition curve should be added

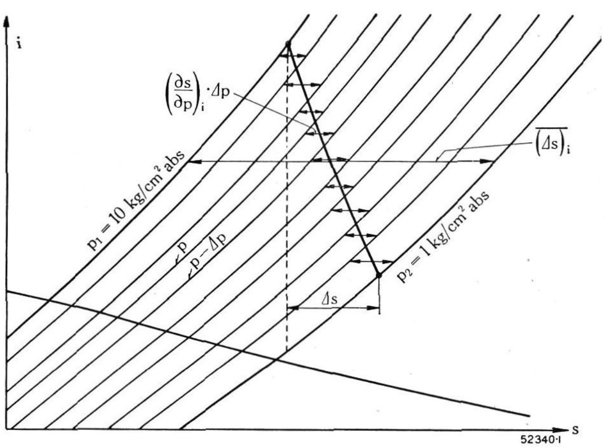


Fig. 3. — Construction of the polytrope in the i-s diagram for steam.

The entropy increase Δs for an expansion in the i-s diagram for vapours is given by the product $(1-\eta_E)$ into the average horizontal distance $\overline{(\Delta s)_i}$ of the neighbouring isobars.

$$\Delta s = (1-\eta_E)\cdot\overline{(\Delta s)_i} \qquad \overline{(\Delta s)_i} = \sum\left(\frac{\partial s}{\partial p}\right)_i\cdot\Delta p$$

together: but a trial will show that the measurement of $\overline{(\Delta s)_i}$ in one place is quite accurate enough, no appreciable correction being found if the curve is subdivided into two or even three parts. Only when the condition curve crosses the saturation line does such a subdivision become advisable.

This proves once again the great value of the i-s chart for turbine calculations, for not only does it enable the total adiabatic efficiency to be determined in the usual manner from the ordinates of the initial and final points, but it also makes possible the direct calculation of the polytropic efficiency from their abscissae.

6. *The polytrope-entropy chart for gases.*

In the case of gases, it is possible to construct an entropy nomogram in which all polytropes (including, therefore, isobars and isothermals) become straight lines and which accordingly may appropriately be called a polytrope-entropy chart, or simply a polytrope chart.

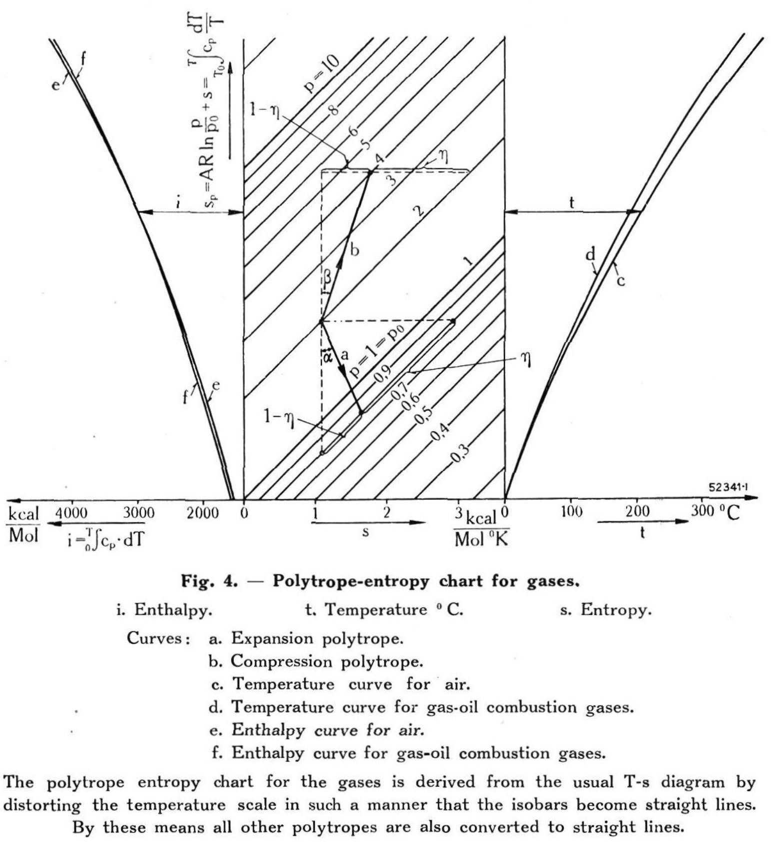


Fig. 4. — Polytrope-entropy chart for gases.

i. Enthalpy. t. Temperature °C. s. Entropy.

Curves: a. Expansion polytrope.
b. Compression polytrope.
c. Temperature curve for air.
d. Temperature curve for gas-oil combustion gases.
e. Enthalpy curve for air.
f. Enthalpy curve for gas-oil combustion gases.

The polytrope entropy chart for the gases is derived from the usual T-s diagram by distorting the temperature scale in such a manner that the isobars become straight lines. By these means all other polytropes are also converted to straight lines.

Fig. 4 shows an excerpt from such a chart in which all quantities are referred to 1 mol. The entropy is plotted as abscissae and the quantity AR log $\frac{p}{p_o} + s$, as ordinates. The latter quantity is nothing other than the entropy s_p along the isobar p_o, that is, $\int_{T_0}^{T} c_p \frac{dT}{T}$.

If the angle α which the adiabatic makes with the expansion polytrope at any particular point is calculated,

$$\tan\alpha = \frac{ds}{-d\left(AR\log\frac{p}{p_o} + s\right)} = \frac{ds}{-AR\frac{dp}{p} - ds}$$

$$= \frac{1}{-\frac{AR}{p}\cdot\frac{dp}{ds} - 1} = \frac{1}{-A\frac{v}{T}\cdot\frac{dp}{ds} - 1}$$

and combining with equation (3)

$$\tan\alpha = \frac{1}{\frac{1}{1-\eta_E} - 1} = \frac{1-\eta_E}{\eta_E} \qquad (6a)$$

and similarly for compression

$$\tan\beta = 1 - \eta_K \qquad (6b)$$

Hence for η_E (or η_K) = constant, α (or β) is also constant. This proves that in the proposed diagram, polytropes are indeed straight lines.

During throttling, $\eta_E = 0$, hence $\alpha = 90^0$, that is, lines of constant i and isothermals are parallel to the abscissae axis. The values assigned to them will depend on the particular characteristics of the gases concerned: on the two sides of the diagram proper there may be drawn curves of temperature and heat content for any gas. If the diagram is used for one particular gas, the isothermals and the lines of constant i may be drawn directly in the diagram.

The family of isobars is the same for all gases. Its spacing is purely logarithmic, and hence intermediate pressures may be interpolated with the aid of a slide rule scale.

A comparison of the proposed entropy chart for gases with the T-s diagram for gases[1] shows the following differences in the T-s diagram: adiabatics, isobars and polytropes are in general curved. Whilst the family of isobars and, — contrary to the polytrope chart — also the constant volume lines may be used for all gases, adiabatics are differently curved according to the gas. Only in the case of one particular gas, for instance air, is it possible to ensure that the adiabatics shall be straight. In the polytrope chart on the other hand, all adiabatics, isobars and polytropes are straight lines and remain the same for all gases.

The polytrope chart, therefore, appears particularly well suited to the study of phenomena where a steady flow is maintained, because in such processes, the changes usually take place along polytropes (in special cases along adiabatics, isobars and throttling lines). Where on the other hand, the changes take place

[1] First proposed by Stodola.

mainly at constant volume such as in explosion processes, the T-s chart might be more convenient. It would, of course, be possible to extend the utility of the polytrope chart by the addition of curves of constant volume and curves of u, but in most cases, it is sufficient that v may be obtained from $\frac{RT}{p}$, and u from i—ART.

In regard to the accuracy with which the characteristic quantities of state may be read off on the diagram, this decreases very rapidly in the T-s chart at low temperatures. Such a disadvantage is avoided by the logarithmic scale of the polytrope chart.

Fig. 4 shows by an example, how the expansion and compression polytropes are constructed. It is interesting to note that the compression polytropes intersect an isothermal so as to form a linear efficiency scale, the expansion polytropes forming a similar scale on an isobar. In other words: if starting from given initial conditions, ***the power of the compressor is kept constant,*** the increase in entropy is exactly proportional to $1-\eta_K$, it is a maximum when $\eta_K = 0$ for the initial isobar, and is zero for the adiabatic. In the case of the turbine the relation takes a somewhat different form: if, again starting from given initial conditions, ***the back-pressure is kept constant,*** the increase in entropy is again proportional to $1-\eta_E$, it is a maximum for pure throttling and zero for the adiabatic.

It is evident that in principle, it would be sufficient to plot only i and t as functions of $AR \log_e \frac{p}{p_0} + s \left(= \int_{T_0}^{T} c_p \cdot \frac{dT}{T} \right)$. The drawing of the purely logarithmically spaced isobar lines could be omitted as the equivalent calculation could readily be carried out on the slide rule although such a process would not be so illustrative. A compression from a pressure p_1 at a temperature t_1 to a pressure p_2 would then require the following procedure:

With t_1 find on t-curve $AR \log_e \frac{p_1}{p_0} + s_1$. To this value there must be added

$$AR \log_e \frac{p_2}{p_1} + \Delta s = AR \log_e \frac{p_2}{p_1} + \frac{1-\eta_K}{\eta_K} AR \log_e \frac{p_2}{p_1}$$

$$= \frac{1}{\eta_K} AR \log_e \frac{p_2}{p_1} = AR \log_e \left(\frac{p_2}{p_1}\right)^{\frac{1}{\eta_K}}$$ [1]

and obtain thus $AR \log_e \frac{p_2}{p_0} + s_2$ from which the corresponding value of t_2 may be read off.

[1] For the meaning of $\frac{1}{\eta_K}$ or η_E as a power index of the pressure ratio see A. Meldahl, "The separation of impeller and diffusor losses in radial blowers", page 205 equation (11).

7. Polytropes for perfect gases with constant specific heat.

If a polytrope chart were to be constructed for a perfect gas with constant specific heat, the temperature scale would be a pure logarithmic one because

$$AR \log \frac{p}{p_0} + s = \int_{T_0}^{T} c_p \frac{dT}{T} = c_p \cdot \log \frac{T}{T_0}$$

If in such a chart (Fig. 5) we were to draw, for instance, a compression polytrope between the pressures

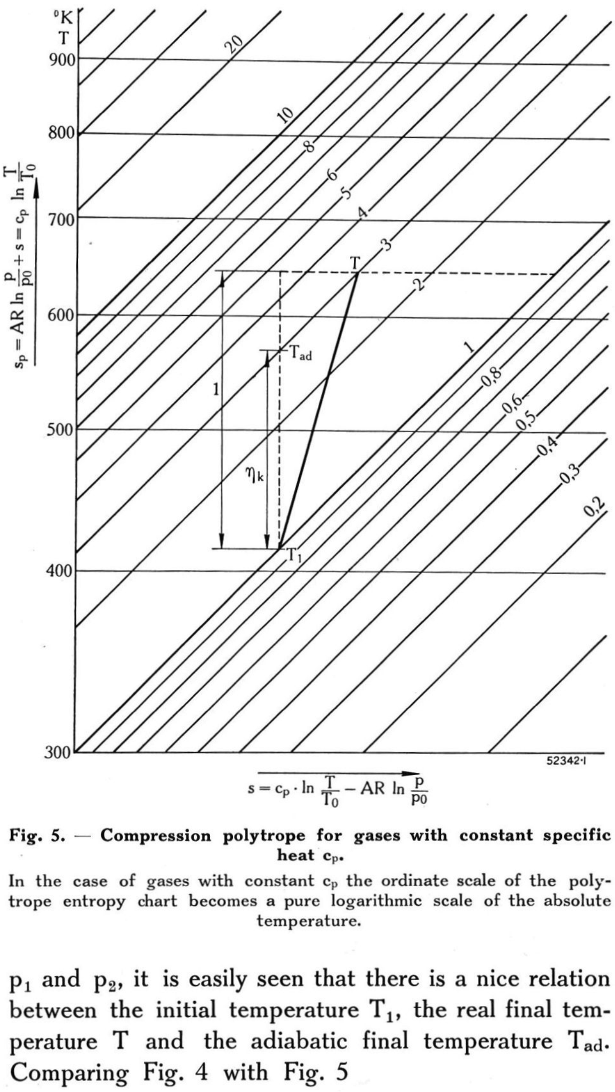


Fig. 5. — Compression polytrope for gases with constant specific heat c_p.

In the case of gases with constant c_p the ordinate scale of the polytrope entropy chart becomes a pure logarithmic scale of the absolute temperature.

p_1 and p_2, it is easily seen that there is a nice relation between the initial temperature T_1, the real final temperature T and the adiabatic final temperature T_{ad}. Comparing Fig. 4 with Fig. 5

$$\frac{c_p \log_e \frac{T}{T_1}}{c_p \log_e \frac{T_{ad}}{T_1}} = \frac{1}{\eta_K} \quad \text{or} \quad \frac{T}{T_1} = \left(\frac{T_{ad}}{T_1}\right)^{\frac{1}{\eta_K}} \qquad (7a)$$

And similarly for expansion

$$\frac{T}{T_1} = \left(\frac{T_{ad}}{T_1}\right)^{\eta_E} \qquad (7b)$$

These relations are very convenient in some calculations. They enable, for instance, when making comparisons to calculate cyclic processes by the temperatures only, without even having to make any assumption as to the value of the adiabatic exponent k. The pressures do not appear at all. Instead of them only adiabatic temperature ratios are used. These can, of course immediately be converted into pressure ratios by the insertion of a suitable k.

8. *Summary.*

A relation is derived for the (polytropic) variation of state in multi-stage heat-insulated turbo-machines by means of which the total increase in entropy Δs may be determined. This enables, the variation of state of both gases and vapours to be readily and quickly drawn in an entropy diagram. If $\overline{(\Delta s)_i}$ denotes the average entropy difference along the i = constant lines between the initial and the final isobar, then Δs is a percentage of $\overline{(\Delta s)_i}$ depending only on the polytropic efficiency η_E or η_K, namely,

$$\Delta s = (1-\eta_E) \cdot \overline{(\Delta s)_i} \text{ for expansion}$$

and $$\Delta s = \frac{1-\eta_K}{\eta_K} \cdot \overline{(\Delta s)_i} \text{ for compression.}$$

Further, an entropy nomogram for gases is proposed (polytrope chart) which has notable advantages over the T-s diagram. In particular, the variation of state in turbo-machines with a constant stage efficiency appears as a straight line, which can be drawn independently of the special characteristics of the gas.

9. *List of symbols.*

Quantities:

p = absolute pressure
v = specific volume
T = absolute temperature
t = temperature ^{0}C
u = internal energy
i = enthalpy
s = entropy
c_p = specific heat at constant pressure
k = adiabatic exponent
R = gas constant
q = heat supplied
A = heat equivalent
η = efficiency
α, β = angles between the adiabatic and the expansion polytrope or the compression polytrope respectively in the polytrope-entropy chart.

Suffixes:

$.._E$ = expansion polytrope
$.._K$ = compression polytrope
$.._{st}$ = turbo-machine stage
$.._o$ = general reference state
$.._1$ = state before turbo-machine
$.._2$ = state after turbo-machine
$.._{ad}$ = theoretical final state on adiabatic.

(MS 771) *O. Zweifel. (Hv.)*

THE GAS-TURBINE LOCOMOTIVE.

Decimal index 625.282—833.8

A gas turbine-electric locomotive for the Swiss Federal Railways is described briefly as an example of a special application of the gas turbine. It burns fuel oil and the power is transmitted electrically to the driving wheels. This is a Type 1 A_o - B_o - A_o 1 express locomotive.

THERE are many railway lines which it does not pay to electrify. The traffic on these lines is not important enough to justify the expenditure of large sums for stationary equipment such as that for contact wire, for feeders or substations. Further such equipment is expensive as regards upkeep, interest and amortization. Train service is infrequent on these lines, but despite, or perhaps just because of this, the weight of the trains is often considerable when the lines are very long. Efforts are made in this way to make the line pay, by putting on heavy express trains with few operators and by utilizing the rolling stock as fully and during as many running hours as possible.

To meet these needs, there are, generally, two well-known methods of traction available:— the steam locomotive and the more recent Diesel-electric locomotive. In its standard form, the steam locomotive with boiler and reciprocating engine has to have coal and a great deal of water and this applies to the very latest designs, as well. About 1 kg of coal and about 9 kg of water are needed per H. P. delivered at the wheel tread. A standard steam locomotive delivering 1000 H. P. at the wheel tread thus consumes in an hour about 1 t of coal and 9 t of water. The result is that its radius of action is restricted. It has to draw a heavy weight of supplies and has to replenish these every few hours. Further it cannot convert into steam any kind of water.

All these drawbacks explain why Diesel-electric locomotives have come into popularity in recent years[1] and are built in units of up to considerable output, in which case electric transmission is used,

[1] According to Railway Age of 4th Jan. 1941 219 steam and 462 Diesel locomotives were ordered in the U.S.A. in the year 1940.

this being, indeed, the best solution. These locomotives require no water except to make up the cooling-water store. Diesel locomotives have an excellent overall efficiency and consume about 250 g of fuel (Diesel oil) and 1 g of lubricating oil per H. P., delivered at the wheel tread, including secondary requirements. In other words, 250 kg of fuel carried are required per 1000 H. P. and per hour, which is about $^1/_{40}$ of what the steam locomotive requires. Thus, the Diesel-electric locomotives can be built for a big radius of action and can carry supplies sufficient for 20 or more running hours. They are quickly put into running order, especially in summer when the cooling water does not need to be heated. Speaking generally, however, these locomotives are heavier, more expensive and require more upkeep than steam locomotives.

Steam locomotives use for fuel coal, or mazout, which has been adopted lately and which is considerably cheaper than Diesel oil.

The gas turbine has now made a locomotive of the kind in question a practical proposition and the Swiss Federal Railways ordered the first gas turbine locomotive in the world from us in 1939. The gas-turbine set works to the so-termed constant-pressure principle (Fig. 1). The set consists of a combustion chamber in which the driving gas is produced by the combustion of oil which is mixed with cooling air, of a reaction turbine with several stages, which converts the energy of the gases into mechanical work, of a compressor the duty of which is to compress the combustion and cooling air to that pressure necessary for doing work. The compressor is of the multi-stage axial type. The delivery pressure amounts to about 3·5 kg/cm² gauge while the temperature of the gas at the turbine inlet is about 550° C. In order to recuperate a part of the heat contained in the exhaust gases and thus to save fuel, a preheater is provided in which the fresh air drawn in is heated by the exhaust gases being expelled.

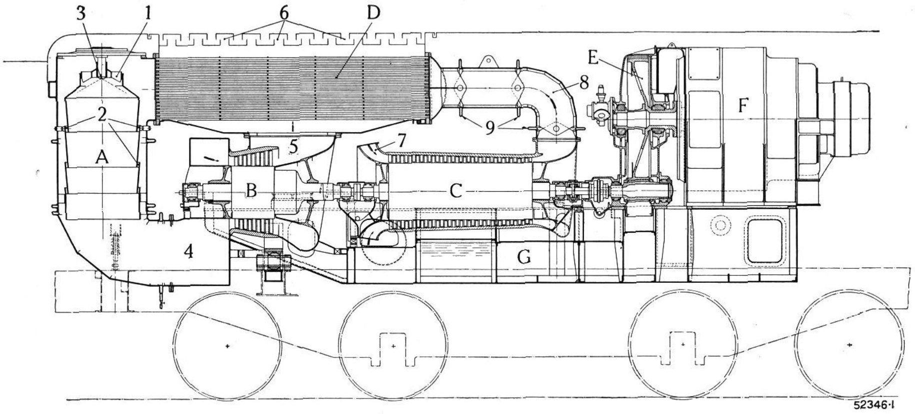


Fig. 1. — Section of a gas turbine for a locomotive.

A. Compressor. B. Gas turbine. C. Combustion chamber. D. Air preheater. E. Reduction gear. F. D.-C. generator. G. Auxiliary frame.

The compressed and preheated air is introduced into the combustion chamber partly as combustion air through the air nozzle ring (with vanes) 1, partly as cooling air through the slits 2, while the fuel is introduced through the injection nozzle 3. Combustion gas and cooling air mix in chamber 4 and form the driving gas proper. The exhaust gases which are still hot enter the preheater at 5 and leave it through the slits 6 in the locomotive roof. Air enters at 7. The air outlet duct 8 is provided with several expansion joints on account of the different expansions of the gas-turbine set and the preheater.

Both kinds of locomotive have piston rods, cranks, excentrics, valves, etc., which are masses in reciprocating movement and are features of all piston engines. They have the inherent disadvantage of being subjected to vibrating phenomena due to resonance, and they demand thorough and ample lubrication.

If, now, it were possible to build a thermal locomotive which had the advantages of the steam and Diesel locomotives without their disadvantages, this new type would, most certainly, be welcomed by a number of railways, especially in countries producing oil and on lines where the trains are heavy, but service infrequent and which it would not be worth electrifying.

The gas turbine set of the locomotive hardly differs from the various ones already built for stationary plants, chiefly as auxiliaries for Velox boilers or chemical plants. The space available on the locomotive placed certain restrictions on the design. This had some effect on the efficiency, which is not quite as high as that of stationary sets, for this reason. Nevertheless, with preheating, we get an efficiency of 17·6 % at the generator coupling (Fig. 2) and this is, already, better than what the best steam locomotive can show although not as good as the efficiency of a Diesel machine. On the other hand, the gas turbine has the advantage of great simplicity and of less cost (Fig. 3). As compared to the steam locomotive, the

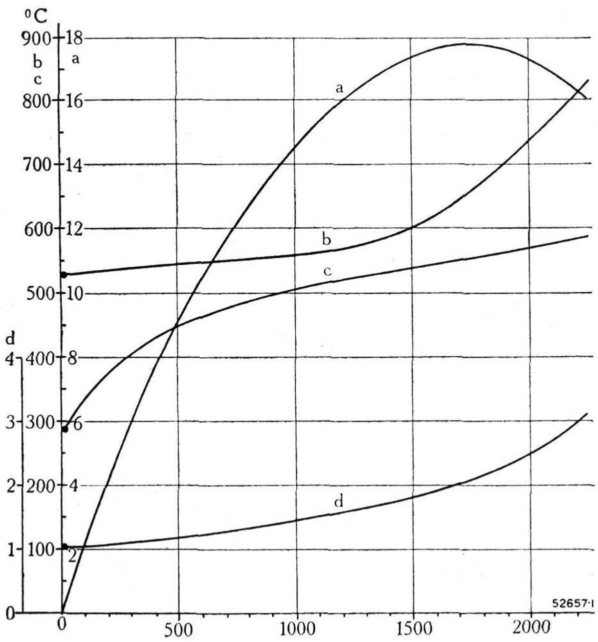


Fig. 2. — Operating magnitudes relative to a gas turbine for traction purposes.

Abcissae: Output in H. P.
Ordinates: a. Thermal efficiency in %.
b. Speed of generator in r. p. m.
c. Gas temperature at turbine inlet ° C.
d. Air pressure after compressor kg/cm² abs.

gas turbine locomotive has the great advantages of being immediately ready for service and of requiring no water. The cooling and essential driving medium of the constant-pressure gas turbine is air. The power generated by the gas turbine can be transmitted to the driving wheels mechanically, hydraulically or electrically. In the present case, electric transmission is used. A main generator and two auxiliary generators are driven through a reduction gear. The two auxiliary generators are for heating and auxiliary requirements, respectively.

The main generator supplies the four d. c. traction motors which each actuate one of the four driving axles of the locomotive through the agency of Brown Boveri spring drives.

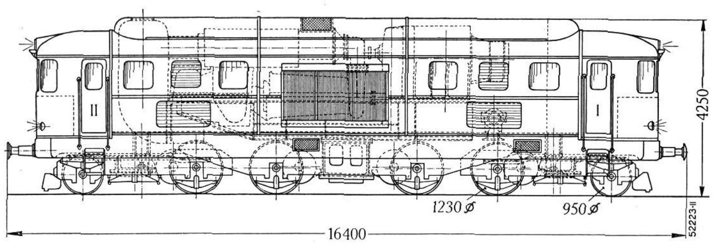
The heating generator delivers the power required to heat the trailers. When the temperature of the outside air drops, there is less work to be done by the air compressor and the useful output of the gas turbine increases we, therefore, get more output in cold weather which is quite sufficient to cover the heating requirements of the train. There being no water used, the locomotive is always ready for work even in the coldest weather.

The third generator delivers D. C. for the auxiliary requirements, such as separate excitation of the main and of the heating generator, power for the compressor and pump motors and to charge a storage battery.

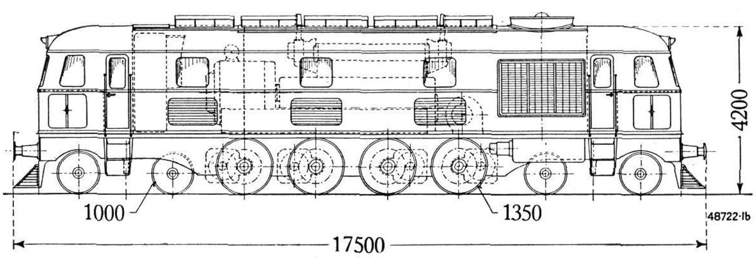
There is a small converter set to supply the lighting of the locomotive and the control current circuit. This set reduces the voltage of the auxiliary generator or of the battery to 36 V, which is the standard lighting voltage of the Swiss Federal Railways. There is not separate lighting storage battery.

The starting of the main set is by electricity, the main generator being run as a motor and supplied with electric power from a Diesel-generator set. After attainment of the speed necessary for ignition, fuel is injected into the combustion chamber and ignited. The set then comes up to rated speed under its own power. With the Diesel-generator set which can deliver a maximum of 110 H. P. the locomotive can run under no load without using the gas turbine at all.

The control system is similar to that of a Diesel-electric locomotive equipped with Brown Boveri control. To operate the gas turbine, it is necessary that it should run at a speed which is a function of the output it delivers. The controller in the cab is the main regulating organ at the disposal of the driver. It adjusts the fuel injection requisite to the momentary power requirement. Simultaneously, the closing position of a sleeve of a speed governor coupled to the gas-



2200 H. P. gas turbine electric locomotive Type 1 A_0-B_0-A_0 1.



2200 H. P. Diesel-electric locomotive Type 2 D_0 2.

Fig. 3. — Comparison between a gas turbine locomotive and a Diesel-electric locomotive.

Weight	Locomotive 1 A_0-B_0-A_0 1 Gas turbine locomotive	Type 2-D_0-2 Diesel-electric locomotive
Mechanical part	37·5 t	50 t
Thermal part	23·7 t	
Diesel plant and access		26 t
Electrical equipment	25·6 t	30·2 t
Stores and equipment	5·2 t	5·8 t
TOTAL WEIGHT in running order	92 t	112 t
Maximum driving axle pressure	16 t	16 t
Runner axle pressure	14 t	12 t
Maximum speed	110 km/h	100 km/h
The gas turbine locomotive is lighter by	20 t.	

turbine shaft is brought to the position corresponding to the speed of the set which is necessary for the compressor if it is to deliver that quantity of air requisite to the amount of fuel injected or desired output. If the momentary speed of the compressor deviates from this desired value, the speed governor varies the field of the generator through the agency of a Brown Boveri servo field-regulator, which then increases or lowers the power input to the generator until the gas turbine has reached the desired speed. The electric transmission permits of the speed of the gas turbine being independent of that of the motors.

The momentary tractive effort of the locomotive (motor current) will always be adjusted, by alteration of the generator excitation, in such a way that it adapts itself to the speed (motor voltage) which corresponds to the output set for on the turbine.

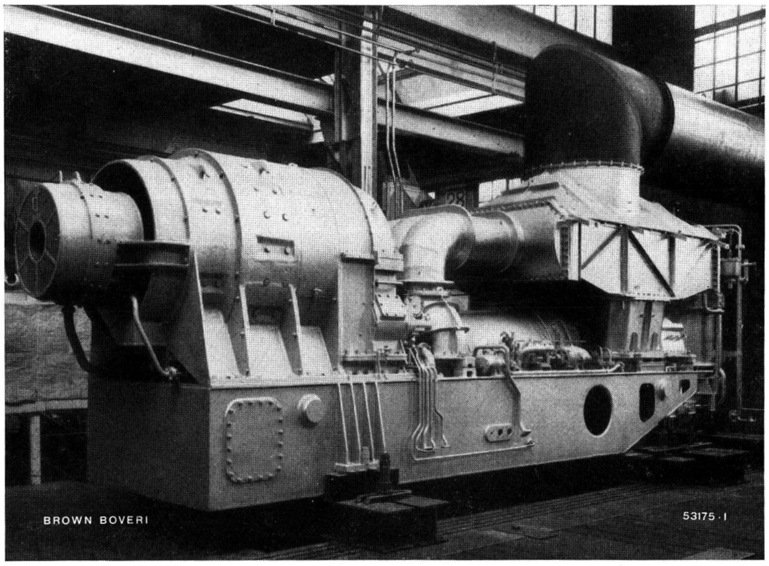


Fig. 4. — Generator set of the gas turbine locomotive on the test bed.
The welded frame of this set contains receptacles for fuel oil and lubricating oil.

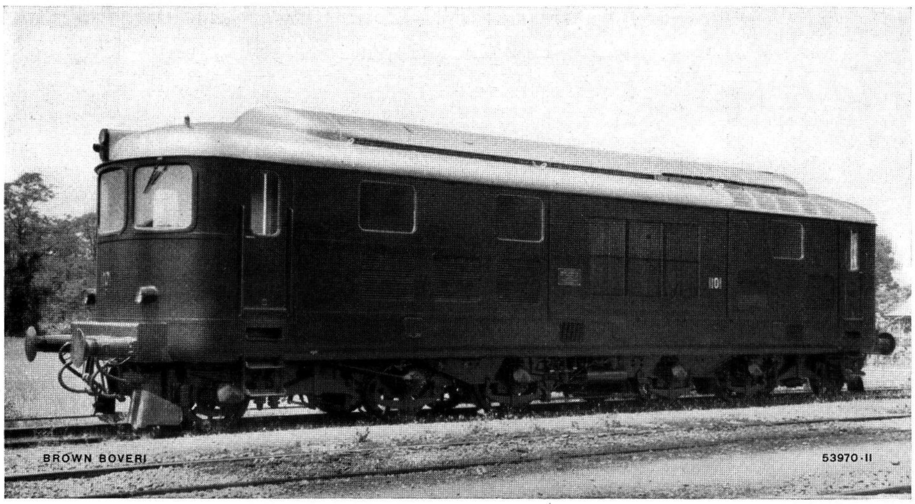


Fig. 5. — The gas turbine locomotive of the Swiss Federal Railways, 2200 H.P., Type $1A_0$-B_0-A_01.

The tractive effort speed data are for half worn tires of 1200 mm rolling diameter and assuming that the auxiliary and heating generators take 263 kW, corresponding to 42 kW delivered by the auxiliary generator and 200 kW by the heating generator. The fuel consumption is about 450 g/H. P. h at the wheel tread. About 900 kg of oil are needed for 2000 H. P. delivered at the wheel tread per hour.

Maximum speed 110 km/h

The main data on the gas turbine-electric locomotive for the Swiss Federal Railways are as follows:

Guaranteed continuous output of the thermal set, measured at generator coupling .	2200 H.P. at 5200/812 r.p.m.
Power available for traction (less the power absorbed by auxiliary services) .	2000 H.P. at 4800/750 r.p.m.
Tractive effort at wheel tread	
during starting . .	13,000 kg from 0 to 26 km/h
during 1 hour . .	7600 kg at 50 km/h
continuous . . .	4840 kg at 78 km/h

Service weight with all stores	92 t	for service on secondary lines (lighter rails).
	93·5 t	for service on main lines (standard rails).
Admissible driving-axle pressure on secondary lines	16 t	
Admissible driving-axle pressure on main lines	18 t	

The total thermal plant (Fig. 4) was mounted in our shops on a common auxiliary frame which also contains the fuel and lubricating-oil tanks.

The whole set with its auxiliary frame was then lodged in the locomotive and secured to the main frame.

Brown Boveri gave the Swiss Locomotive and Machine Works, Winterthur (Switzerland) the order for the mechanical part. The locomotive was ready this summer, it is to be on trial with the Swiss Federal Railways for one year and then taken over by them if it fulfils the conditions specified. It is to be used on those sections of the Swiss Federal Railway system which have not been electrified, or on main lines as a stand-by if current fails.

We think that the gas turbine and the gas turbine-electric locomotive have a big future. By placing this order, the Swiss Federal Railways showed vision in providing the opportunity of building this, the first, gas-turbine locomotive; they did a service to the export business of the country by encouraging the building of a type of machine which is peculiarly suited to conditions prevalent in certain other countries.

(MS 768) *E. Schroeder. (Mo.)*

BLAST GENERATION AND BLAST-HEATING IN IRON WORKS.

(A COMPARATIVE STUDY.)

Decimal index 669.162.23 : 621.438

It will be shown how supercharging and gas turbines can be turned to useful account in iron works especially in blast-furnace plants, thanks to fuels in gaseous form being available. A summary description is given of various applications, as, for example, of the Velox boiler for generating the steam, of the "supercharged" recuperator for blast heating and of the gas turbine to drive the blast blower. The different layouts are shown diagrammatically. Comparative calculations supply data on the economic side of the new plants and of those used up till to-day.

BLAST-FURNACE gas is chiefly used in iron works to generate heat and power. Therefore iron works offer a varied field of application for the supercharging process and for gas turbines. The object of this article is to show what use is made of the supercharging process and of gas turbines in iron works for the purpose of *blast generation* and *blast heating* and also what consumption of heat or gas result from the different designs and processes employed.

The advantages of supercharging and of the gas turbine are not solely on the fuel side. It is still more important that, thanks to supercharging and to gas turbines, it becomes possible to make all the parts of smaller dimensions, so that they demand less building material, take up a smaller area and comprise smaller buildings. Thus it becomes feasible to bring the whole blast-generating plant close up to the blast furnace itself and to operate the blast-furnace and the blast generating and heating plant as a compact unit.

Its unparalleled simplicity makes the gas turbine especially suitable for use in iron works. As compared to the ordinary gas engine, the most obvious advantage it has is that of considerably lower first costs while, as compared to a steam plant, we have the great advantage of the elimination of a boiler, of a condenser plant and of the whole problem of getting water and treating it.

For heating the blast, the supercharging process is of especial importance. Both regenerators (Cowper stoves) as well as recuperators (steel blast heaters) can be built to incorporate supercharging. The future certainly lies with the steel recuperator. It allows of more uniform operation and has a higher efficiency than a Cowper stove; for the time being, however, it is limited to low blast temperatures. It is very suitable for supercharging, but must be specially built for this on account of the higher hot-gas pressures. The design of the Velox steam generator can be taken as a model for the recuperator. As in the latter, the hot gases are made to flow parallel to the tubes in order to avoid pressure being exerted on one side of the tubes which are at a high temperature and, therefore, not able to withstand high pressures. If the pressure of the hot gases is maintained about equal to the pressure of the blast air, practically all stressing due to inner pressure is eliminated.

To generate the blast and the driving medium of the gas turbine, one and the same blower can be used. As the amount of air used for the blast is much smaller than that to be delivered for combustion and cooling purposes to the gas turbine, the fluctuations in pressure and amount of blast, which are inherent to blast furnace operation, have but little effect on the behaviour of the common blower. For these reasons the multi-stage axial compressor with its far higher efficiency replaces advantageously the centrifugal blower used in iron works up till now exclusively.

Really compact plants result from combining together not only the blast blower and the gas turbine blower but the gas-turbine and the blast heater as well[1]. The gas turbine and the blast heater have then got a common combustion chamber. The blast heater is, then, a part of the gas turbine, its exhaust gases are driving gases for the turbine. As, in this way, the total exhaust-gas loss from the blast heating is eliminated, the combination in question is very advantageous. If the exhaust gases from the blast heater have a temperature which is still higher than is admissible for gases to drive the gas turbine, they are cooled by air tapped directly from the blower.

As highest hot-gas temperature for the steel blast heater 1100 °C is assumed and 750 °C as highest blast temperature. For poor grade ore this blast temperature suffices and, indeed, is not attained in many cases. For brickwork stoves the hot blast temperatures are usually higher. For purposes of comparison, however, a temperature of 750 °C is taken for the Cowper stoves in the following table. — It can be assumed that in future higher temperatures combined with satisfactorily long life will become possible for steel blast heaters after the necessary experience has been gained and further progress achieved in the manufacture of heat-proof steel.

The following assumptions are at the base of the comparative calculations the results of which are given here.

Fuel:	Blast-furnace gas 906 kcal/Nm^3 lower calorific value.
	Theoretical quantity of air 0·705 m^3/m^3 blast-furnace gas.
Blast quantity:	80 000 Nm^3 or 103 200 kg/h.
Blast pressure:	at furnace 2·2 kg/cm^2 abs.
Hot blast-temperature:	750 °C in steel blast heater constant, in Cowper stove average value.

[1] DRP 694668 and other patents.

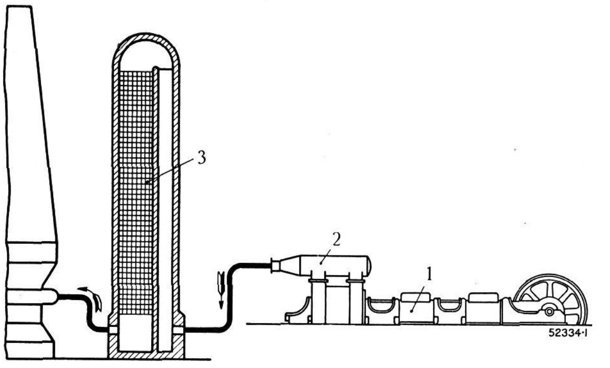
All calculations are made for the best point (nearly full load). Deviations at partial load are lowest for a plant with Velox and are biggest for plants with gas engines (admitting that these operate at partial load).

The examples now given show what the heat consumption is, in kcal, for each 1 Nm^3 of blast compressed to 2·2 kg/cm^2 abs and heated up to 750 °C. Assuming that the fraction of blast-furnace gas utilized in a gas-engine and Cowper-stove plant is about 30 to 32 % of the whole amount of gas generated, which is approximately exact, we get for a good steam plant with ordinary boilers under the

above assumptions . . .	about	30	to	32	%
For a plant with Velox boilers	,,	29·5	,,	31	%
For a plant with gas turbine	,,	29·5	,,	31	%
For a plant with gas turbine and common combustion chamber for gas turbine and blast heater	,,	25	,,	26·5	%

Apart from the simplification from the mechanical point of view, the economic superiority of the gas turbine blast-heater operation is obvious.

As regards first cost of plant, we will only say here that the gas turbine plant is about equal to the steam plant, both being considerably cheaper than a gas-engine Cowper-stove plant.



Gas engine (1)

with reciprocating blower (2) and Cowper stove (3). The gas machine can also be equipped with an exhaust-gas boiler in which steam is generated from the heat in the exhaust gas.

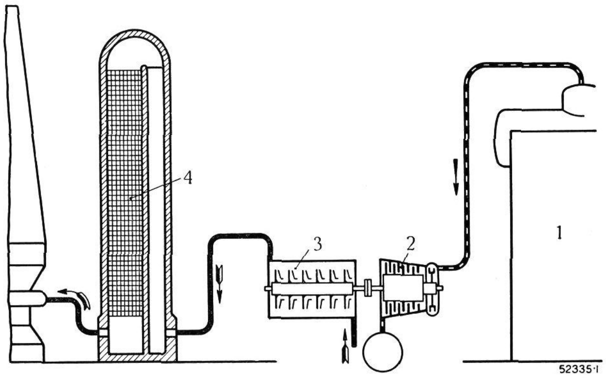
Blast generation:

Amount of air	108,500 kg including 5 % loss
Pressure ratio of compressor . .	2·225
Work of compression	15·65 kcal/kg referred to the isotherm 15 °C
Efficiency of compression . . .	0·66 including mechanical and leakage losses
Output of the gas engine . . .	3 000 kW

	(a) without exhaust-gas boiler	(b) with exhaust-gas boiler
Efficiency of gas engine plant Monthly average	0·22	0·26
Heat-consumption of gas-engine .	11,700,000	9,900,000 kcal/h

Blast heating:—

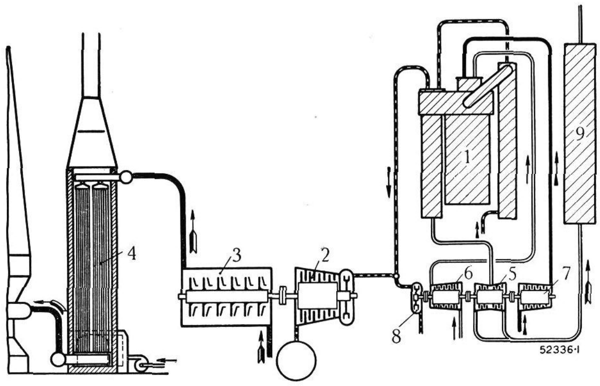
Amount of air	103,500 kg/h	
Cold blast	90 °C	
Hot blast	average 750 °C	
Efficiency of Cowper stove . .	0·78	
Heat consumption for heating blast .	22,320,000	22,320,000 kcal/h
Total	34,020,000	32,220,000 kcal/h
or per Nm^3 blast	**425**	**402·5 kcal.**



Steam plant.

Composed of a *water-tube boiler* (1) heated by blast-furnace gas and a steam turbine (2). Blast generated in a turbo-blower (3). Blast heated in a Cowper stove (4) or steel blast heater, fired under atmospheric pressure and with cooling by returned gases so as to bring down the temperature of the heating gases without loss.

Blast generation:—	(a) with Cowper stove	(b) with steel blast heater
Amount of air	108,500	103,500 kg/h
Pressure ratio of compressor	2·225	2·245
Work of compression	17·80	18·20kcal/kg
Efficiency of compression	0·755	referred to adiabatic *compression*
Output of the steam turbine	2970	2900 kW
Steam consumption 38 kg/cm² abs, 410° C at boiler, 35 kg/cm² abs, 400° C at turbine	4·43	kg/kWh incl. auxiliaries
Efficiency of boiler	0·815	incl. all auxiliaries and boiler feed pump
Heat consumption of blast-generation	11,800,000	11,500,000kcal/h
Blast heating:—		
Amount of air	103 500	103,500 kg/h
Cold blast temperature	90	100° C
Hot blast temperature	750	750° C
Efficiency of blast heater	0·78	0·82 incl. fans.
Heat consumption for heating blast	22,320,000	20,980,000kcal/h
Total	34,120,000	32,480,000
or per Nm³ blast	**426·5**	**406** kcal.

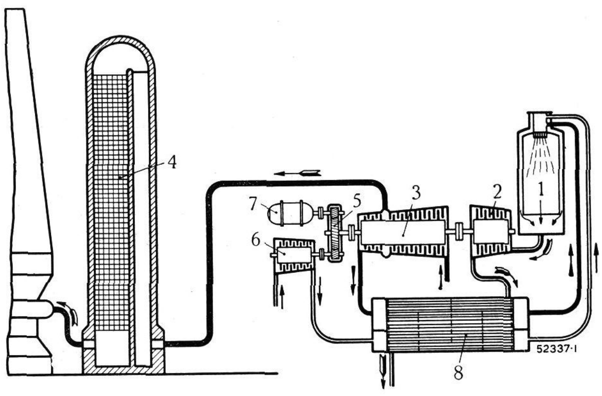


Steam plant.

Composed of a *Velox boiler* (1) fired by blast-furnace gas and of a steam turbine (2). Blast generation by turbo blower (3). Blast heating in a Cowper stove or in a steel blast heater (4), fired under atmospheric pressure and with cooling by returned gases so as to bring down the temperature of the heating gases without loss.

The Velox boiler is charged by the blowers (6) and (7) driven by the gas turbine (5). The steam turbine (8) is utilized for starting and regulating the charging set. (9) is the economizer.

Blast generation:—	(a) with Cowper stove	(b) with steel blast heater
Amount of air	108,500	103,500 kg/h
Pressure ratio of compressor	2·225	2·245
Work of compression	17·80	18·20kcal/kg
Efficiency of compression	0·755	referred to adiabatic compression.
Output of steam turbine	2970	2900 kW
Steam consumption:— 38 kg/cm² abs, 410° C at boiler, 35 kg/cm² abs, 400° C at turbine	4·43	kg/kWh incl. auxiliaries
Efficiency of boiler	0·885	incl. all auxiliaries and *boiler feed pump*
Heat consumption for blast generation	10,860,000	10,580,000kcal/h
Blast heating:—		
Amount of air	103,500	103,500 kg/h
Cold-blast temperature	90	100° C
Hot-blast temperature	750	750° C
Efficiency of blast heater	0·78	0·82 incl. fans.
Heat consumption for heating blast	22,320,000	20,980,000kcal/h
Total	33,180,000	31,560,000kcal/h
or per Nm³ blast	**414·5**	**394·5**kcal



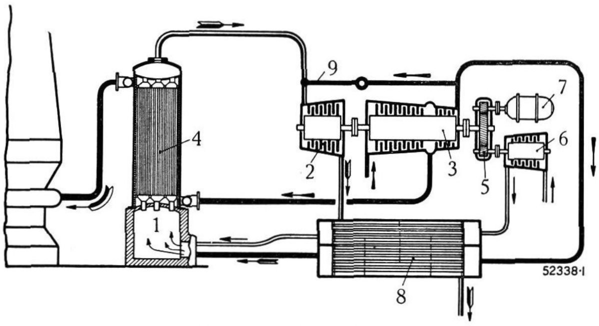
Gas-turbine plant.

Compose of a combustion chamber (1) fired by blast-furnace gas and of the gas turbine proper (2). Blast generated in axial blower (3), which delivers the air for the gas turbine combustion and the air for the gas-turbine cooling simultaneously. Blast heating in a Cowper stove (4) or in a steel blast heater, fired under atmospheric pressure. Apart from the air blower, the gas turbine drives the gas blower (6) *through a reduction gear* (5). (7) is the starting motor.

The exhaust gases from the gas turbine are used in a heat exchanger (8) to preheat the gas-turbine driving medium.

Blast generation:—	(a) with Cowper stove	(b) with steel blast heater
Amount of air	108,500	103,500 kg/h
Pressure ratio of compressor	2·225	2·245
Work of compression	17·80	18·20 kcal/kg
Efficiency of compression	0·825	0·825
Compression output for blast alone	2720	2650 kW
Pressure of driving gases	3·100	3·100 kg/cm³abs
Temperature of driving gas	550	550° C
Total output of gas turbine	9700	9450 kW
Size of preheater[1]	3000	3000 m²
Heat consumption for blast generation	10,800,000	10,520,000 kcal/h
Blast heating:—		
Amount of air	103,500	103,500 kg/h
Cold blast temperature	90	100° C
Hot blast temperature	750	750° C
Efficiency of blast preheater	0·78	0·82 inc. fans.
Heat consumption for heating blast	22,320,000	20,980,000 kcal/h
Total	33,120,000	31,500,000 kcal/h
or per Nm³ blast	**414·5**	**394** kcal
Without preheating the blast gas		
Size of preheater (for air alone)[1]	2500	2500 m³
per Nm³ blast	**426**	**406** kcal

[1] By enlarging the preheater, the heat consumption can be reduced.



Gas-turbine plant.

Composed of a combustion chamber (1) fired by blast-furnace gas and of the gas turbine proper (2). Blast generation by axial blower (3) which also delivers air for the gas-turbine combustion and air for the gas-turbine cooling. *Blast generation* takes place in a *supercharged recuperator* (4) which is heated by the same gases which form the driving medium of the gas turbine.

Apart from the air blower, the gas turbine drives the gas blower (6) through a reduction gear (5). (7) is the starting motor.

The exhaust gases from the gas turbine are used in a heat exchanger (8) to preheat the gas-turbine driving medium.

The gases expelled from the blast heater (4) are too hot for the gas turbine. Therefore, a connecting pipe (9) is provided through which cold air or also preheated air can be passed, to be mixed with the hot gases. This mixture allows of maintaining exactly the proper driving gas temperature, so that it is admissible to go as high as 600° C with this gas.

	Supercharged recuperator	
Temperature before gas turbine	(a) 550	(b) 600 ° C
Blast generation:—		
Amount of air	103,200	103,200 kg/h
Pressure ratio of compressor	2·38	2·38
Work of compression	19·5	19·5 kcal/kg
Compressor efficiency	0·825	0·825
Compression output for blast alone	2840	2840 kW
Pressure of driving gas	3·50	4·00 kg/cm² abs
Temperature of driving gas	550	600 ° C
Total output of gas turbine	10,500	9800 kW
Size of preheater	3200	3200 m²
Heat consumption for blast generation	11,450,000	10,000,000 kcal/h
Blast-heating:—		
Amount of air	103,200	103,200 kg/h
Cold blast temperature	110	111 ° C
Hot blast temperature	750	750 ° C
Efficiency of blast-heater	0·980	0·980
Heat consumption for blast heating	17,100,000	17,000,000 kcal/h
Total	28,550,000	27,000,000 kcal/h
or per Nm³ blast	357	337·5 kcal/h

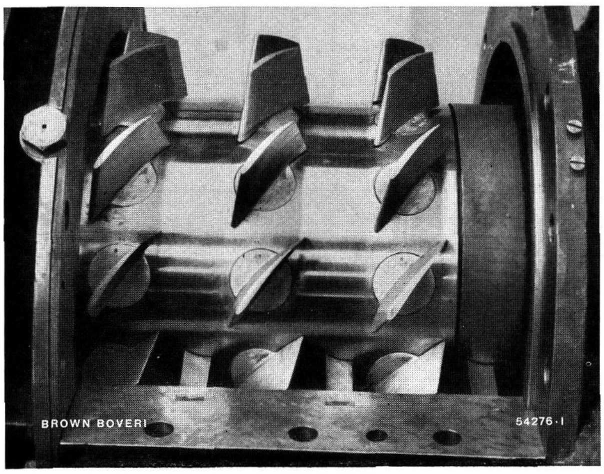
(MS 778) *Dr. W. G. Noack. (Mo.)*

BRIEF BUT INTERESTING

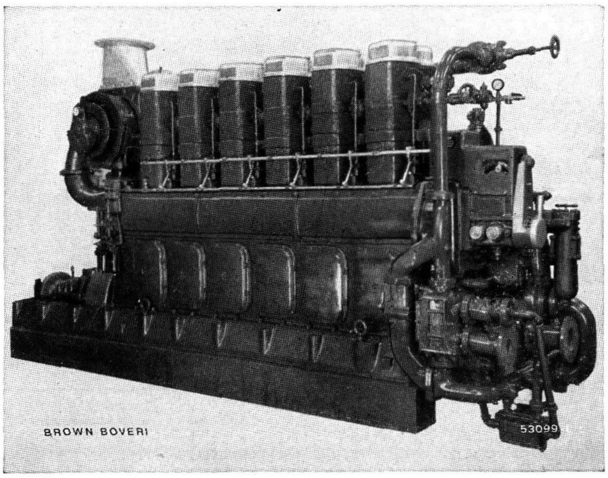
THE first charging sets with gas-turbine drive delivered by Brown Boveri, which could no longer be spoken of as test units, date from the year 1926. All sets delivered at that time are still in service to-day. A short time ago we heard from a client, who took delivery of a charging set in the year 1931, and who to-day has three such sets in his station, that the first set ran more than 14,000 hours, without at any time requiring maintenance worth mentioning or replacement of ball or roller bearings.

The large Diesel locomotive of the Rumanian State Railways, in which two charged Sulzer Diesel engines, each of 2200 E. H. P., are installed, has now covered a distance of over 210,000 km. The four turbo-supercharging sets supplied by our firm have always run perfectly. Up to the present time not one single important part has had to be replaced.

Axial blowers with fixed blading can only be used for limited ranges of delivery volumes. With adjustable moving blades, such a blower could be perfectly regulated, and it is then capable of delivering practically every volume and every pressure from zero up to the maximum value without change of speed. A three-stage, high-speed blower with adjustable moving blades has been built for test purposes, and runs satisfactorily. The illustration on the left shows the blower on the test-bed. It delivers normally 3 m³/s at a pressure of 1600 mm water column, and runs at 7800 r. p. m. The blade angle can be adjusted during service between 0° and 45°. A motor running at 1500 r. p. m. drives the blower over a pair of gears built into the end-shield.



Six-cylinder four-stroke Diesel engine built by the Enterprise Corporation in San Francisco with Brown Boveri exhaust-gas turbine and charging blower.



Two such engines are installed in a tug, which operates at the mouth of the Columbia River in North America. The output of each motor is 700 H. P., which is increased by supercharging to 1000 H. P. at 650 r. p. m.

We have built, up to the present, more than 1500 such charging sets, the Diesel engines supercharged by these sets having a total output of over 1·6 million H.P.

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