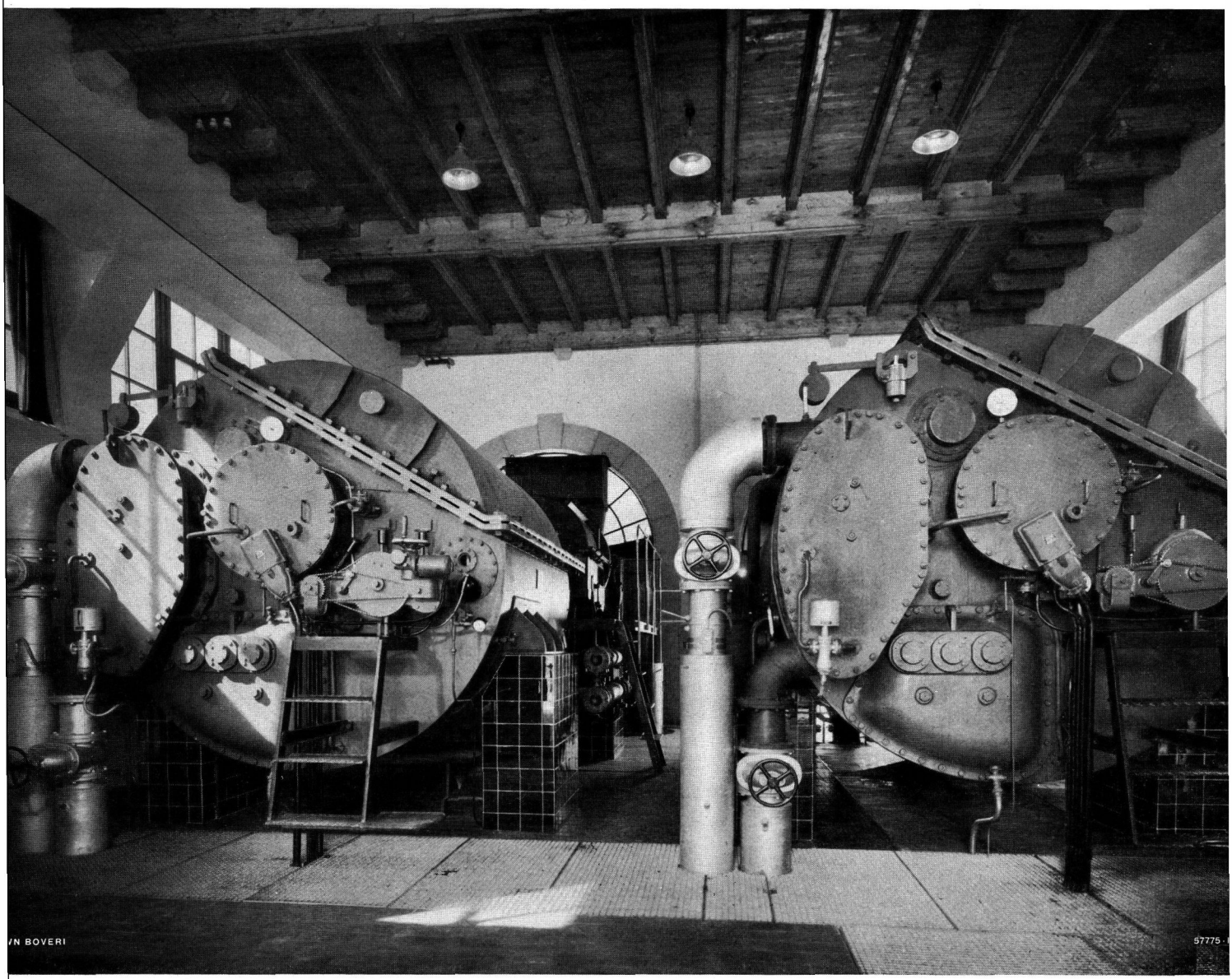
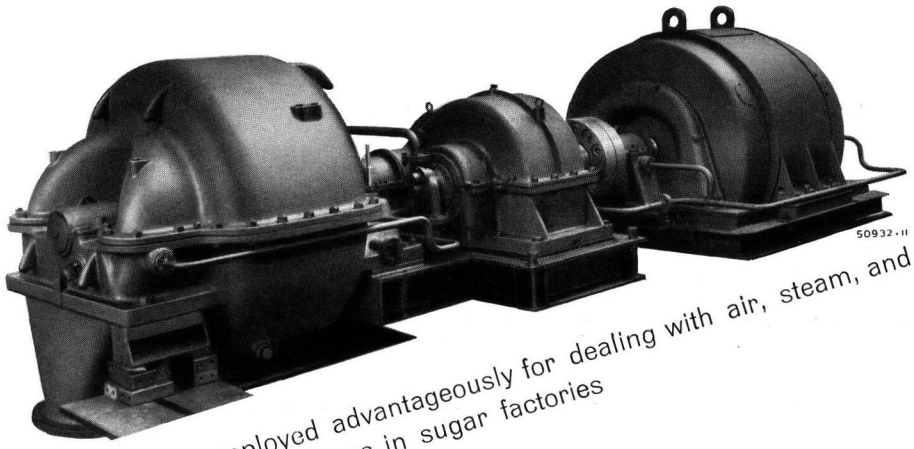
THE BROWN BOVERI REVIEW



Two heat pumps, each rated 1—1·7 million kcal/h,
installed at the works of the Steckborn Kunstseide A.-G., Steckborn (Switzerland)
for heating purposes.

Due to the big saving in fuel realized with heat pumps this class of equipment is meeting with ever-increasing interest in large industrial works.

OL. XXX JULY/AUGUST, 1943 Nos. 7/8



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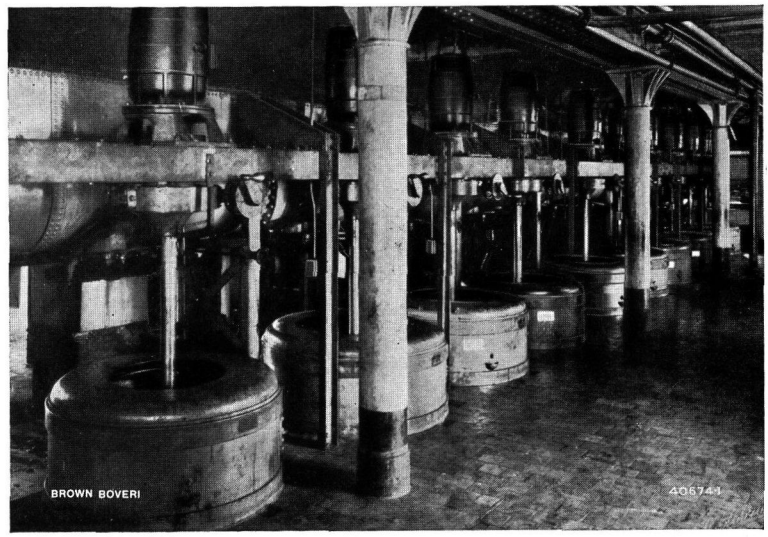
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THE BROWN BOVERI REVIEW

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VOL. XXX JULY/AUGUST, 1943 Nos. 7/8

The Brown Boveri Review appears monthly. — Reproduction of articles or illustrations is permitted subject to full acknowledgment.

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THE POST-WAR STEAM POWER PLANT.

Decimal Index 621.1

Higher coal prices may be expected in the post-war period. These will call for steam power plants of higher efficiency and will enable considerably higher prices paid for such machine to be amortized in a short time by the saving in coal achieved. The present article suggests where and how such improvements can be obtained. Reference is made in particular to a "hot-steam" turbine plant operating at temperatures up to 600° C, with a separately fired combined superimposed superheater and intermediate reheater working on the Velox principle. With such plants, overall efficiencies from fuel to terminal kW of over 35 % and heat consumptions under 2450 kcal/kwh may be attained.

INTRODUCTION.

UPON glancing at the title, it might be asked: How can anything useful be said about what will happen after the war, as long as it is not known when, and especially how, it will end. A few reflections, a review of the conditions of the post-war period of the last world-war, and some statistics taken from that period, provide us, however, with valuable pointers, at least as to the direction in which events may be expected to move.

As generally known, the total costs in the case of a steam power plant consist on the one hand, of the running costs, covering mainly wages, fuel, and lubricating costs and on the other of the fixed charges, such as interest and amortization on the capital outlay, taxes, etc. Depending on the number of service hours and the average load, either the one or the other will predominate in the total yearly expenses. For this reason, the choice will fall in the case of a small number of operating hours and of a low load factor, on the cheapest possible type of machine, without regard to its efficiency, in order to reduce as far as possible, the fixed costs which represent the greater part of the total, and compared with which, the cost of fuel is negligible. On the other hand, in the case of a large number of operating hours and of a high load-factor, the choice will fall, even if its first cost is high, on a machine having a good efficiency, in order to reduce by this means the fuel costs which now predominate.

Other conditions being the same, the higher the price of fuel, the fewer will be the number of service hours at which the operating costs begin to exceed the fixed charges, and the more important becomes the question of improving the efficiency when renovating or extending thermal power plants.

Fig. 1 illustrates the coal supply conditions for Switzerland from 1910 to 1942. The diagram shows: the quantity of coal imported in millions of tons, its import value in millions of francs, and the price of coal delivered Basle, duty excluded, over the stated period. The large peak in the total import value and in the price per ton, as well as the considerable drop in the import quantity during the war and post-war years 1916 to 1923, are particularly noticeable.

Considering first the price per ton, we see that it rose from Fr. 25.— per ton in the year 1912 up to a maximum of Fr. 200.— in the year 1920, that is, two years after the end of the war, to fall rapidly within the next two years to Fr. 65.— and thereafter more slowly until the year 1934. In that year, the pre-war price and even slightly lower prices were again attained. The excess of the import value above the

peace-time cost is worth noting. It amounts to no less than 1100 million francs. This amount may be looked upon as a part of Switzerland's contribution to the cost of the war.

Now it is interesting to consider the form of the three curves after, and immediately prior to, the outbreak of the last war, and to compare them with corresponding variations at the outbreak of the present war. For this purpose, the curve showing the rise in price of coal after the outbreak of the war in 1914 has been reproduced as from 1939 in Fig. 1. The comparison shows that just as in the case of the first world war, a rise in price took place at the outbreak of the second one, however with the difference, that this time, probably due to the lessons of the first war, the rise started sooner, and was also steeper. Apart from this, the variation in price per ton follows a similar curve. The import quantity also, was wisely increased already in 1937 to attain a maximum in the year 1939 before the rapid drop to its present level took place.

Instead of displacing the curve of coal price to the dotted position on the right, the same result can be achieved by displacing the time base to the left, as has been done along the abscissæ axis of the diagram. The displacement has been made such that the price of coal of the year 1940 coincides with the same price during the last world war. The year 1940 thus falls in the middle of the year 1917, and it may be accordingly expected that the subsequent variations will be along a similar curve.

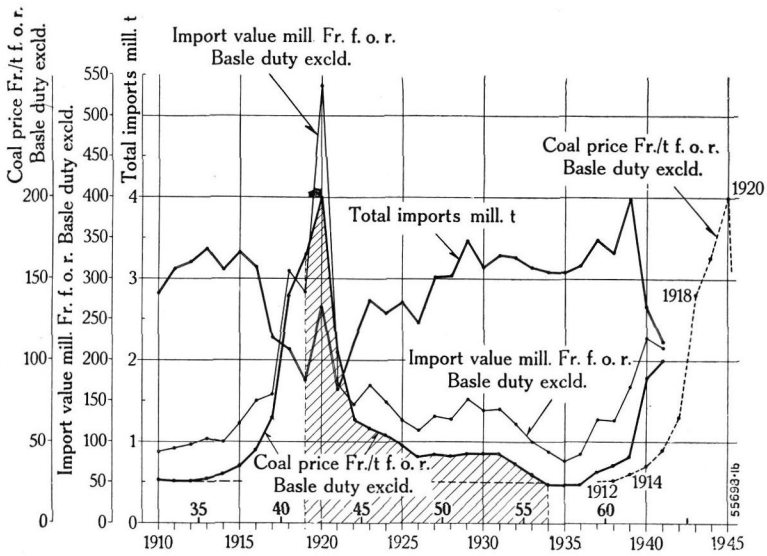


Fig. 1. — The evolution of Swiss coal imports.

Note the quick increase in price at the outbreak of the war, the maximum being attained only two years after the end of the war. This is followed by a slow fall returning to peace time price, fifteen years after the outbreak of the war.

In regard to the anticipated trend of coal and oil prices, most authorities agree that after the second world war, fuel prices will never again fall to their former levels, because considerably more coal will be used for the manufacture of various products, such as petrol, oil, artificial rubber, and chemical goods. This assumes a far-reaching autarchy of European states or groups of states. Other minds are of the opinion that, even for a "United States of Europe", absolute autarchy is neither desirable nor in the long run possible, as these states will wish to export, and therefore will be in a position to import oil and other natural products which are much cheaper than the synthetic ones.

All these considerations show that, whatever may happen, it may be assumed with a high degree of certainty that the post-war price of coal will not be lower than the pre-war one, and that, if it ever does attain the pre-war level, it will not do so until several years after the end of the war. If it remains permanently higher, then the conclusions deduced in the following are even more justified.

Let us assume that the trend of the prices one year after the present war will be the same as that one year after the first world war, i. e., the same as for the year 1919, and that from this moment onward, the price of coal will follow the same curve as after the first world war, the hatched surface 1919—1934 in the diagram multiplied by the coal consumption of a machine in a year will hence represent the fuel costs during the post-war period.

A simple calculation will then show what sums can be saved yearly due to a certain percentage improvement in efficiency, to cover the amortization and interest charges on the capital which might be required to achieve this improvement in efficiency.

Thus, for example, investigation shows that the saving in coal due to a 10 % improvement in efficiency of a steam power plant will enable an increase in cost of 30 % to be written off in three years, and one of 50 % to be written off in five years, from which it is clear:

(a) that the post-war period will call for high-quality machines, consuming as little fuel as possible.

(b) that even appreciable extra costs will be recovered in very short periods.

Let us now consider what are the possibilities of improvement of the different types of prime movers, and upon which of the lines of development followed up to the present these improvements are likely to take place. Taking the machines in the chronological order of their appearance, we come first to

THE STEAM ENGINE.

The steam engine has had to cede the greater part of its former field of application to the turbine, and even its remaining market is not uncontested. Nevertheless, this is very much larger than is generally appreciated, because, partly owing to the natural conservatism of shipowners and shipbuilders on the one hand, and partly because of their greater simplicity enabling the use of relatively unskilled personnel on the other, steam engines still find application in the great majority of ship's drives for powers up to 4000 H.P.

Already at an early date, makers began to adapt steam engines to the rise in temperatures and pressures which has gone on steadily since Watt's time, by connecting two or more cylinders in series; i. e. by resorting to double-, triple- or even quadruple-expansion engines, quite respectable pressures can be handled with high efficiency.

On the other hand, due to the rapidly increasing volume of the steam and due to the condensation taking place in the cylinders, a limit is soon reached to the vacuum which can be utilized with a steam engine, even by paralleling low-pressure cylinders or by adopting the uniflow principle.

One way of avoiding this difficulty consists in the addition of an exhaust-steam turbine placed after the steam engine. As this measure can be applied to existing installations, interesting developments in this direction may be expected in the post-war period. The appearance of such an exhaust-steam installation is shown by Figs. 2 and 3, which illustrate in elevation and in plan, a type of plant such as has been supplied for the steamships "Blitar", "Ammon" and "Amasis". When converting such an installation, a reduction gear is interposed between the reciprocating engine and the propeller shaft, which transmits the extra power of the exhaust turbine through several reduction stages to the main shaft. The improvement in steam consumption of 25—30 % thus obtained, enables a corresponding increase of power without having to increase the boiler capacity. A noteworthy characteristic of this Brown Boveri exhaust-steam system lies in the fact that the exhaust turbine is connected to the reciprocating engine not by a hydraulic but by a spring coupling; thus the crank effort diagram is improved to such an extent by the inertia of the rapidly rotating mass of the turbine that it is, as a rule, not necessary to replace the ship's shaft to transmit the increased power. These conditions are illustrated in Fig. 4, which shows how the crank effort diagram of a reciprocating engine is modified and smoothed out by the addition of a Brown Boveri exhaust turbine.

Another problem with the steam engine is the difficulty of using high steam temperatures, due to the necessity of lubricating the cylinders and pistons, and due to the instability of lubricating oils at high temperatures. For this reason, the great majority of steam engines still operate with saturated or only slightly superheated steam. It has been suggested to employ higher superheats for the purpose of reheating the steam between cylinders and thus communicate some of the superheat energy of the live steam to the partly expanded steam before entering the high-pressure cylinder. Fig. 5 shows a plant on these lines built according to a suggestion of professor G. Bauer of Hamburg[1]. It is of course possible to place in front of the steam engine of such an installation a superheated steam turbine in accordance with the arrangement shown in Fig. 6, which is particularly readily done, when, as in the case illustrated, an exhaust turbine and hence, a gear, is present to which the superheated steam turbine can be coupled. It may, however, then well be asked, what is the object of the steam engine, as due to the many additions the simplicity which made for easy attendance, has been lost.

How such an extension can advantageously be made to an existing marine boiler plant is shown by Figs. 7

[1] Zeitschrift des Vereins Deutscher Ingenieure, 26th July, 1937.

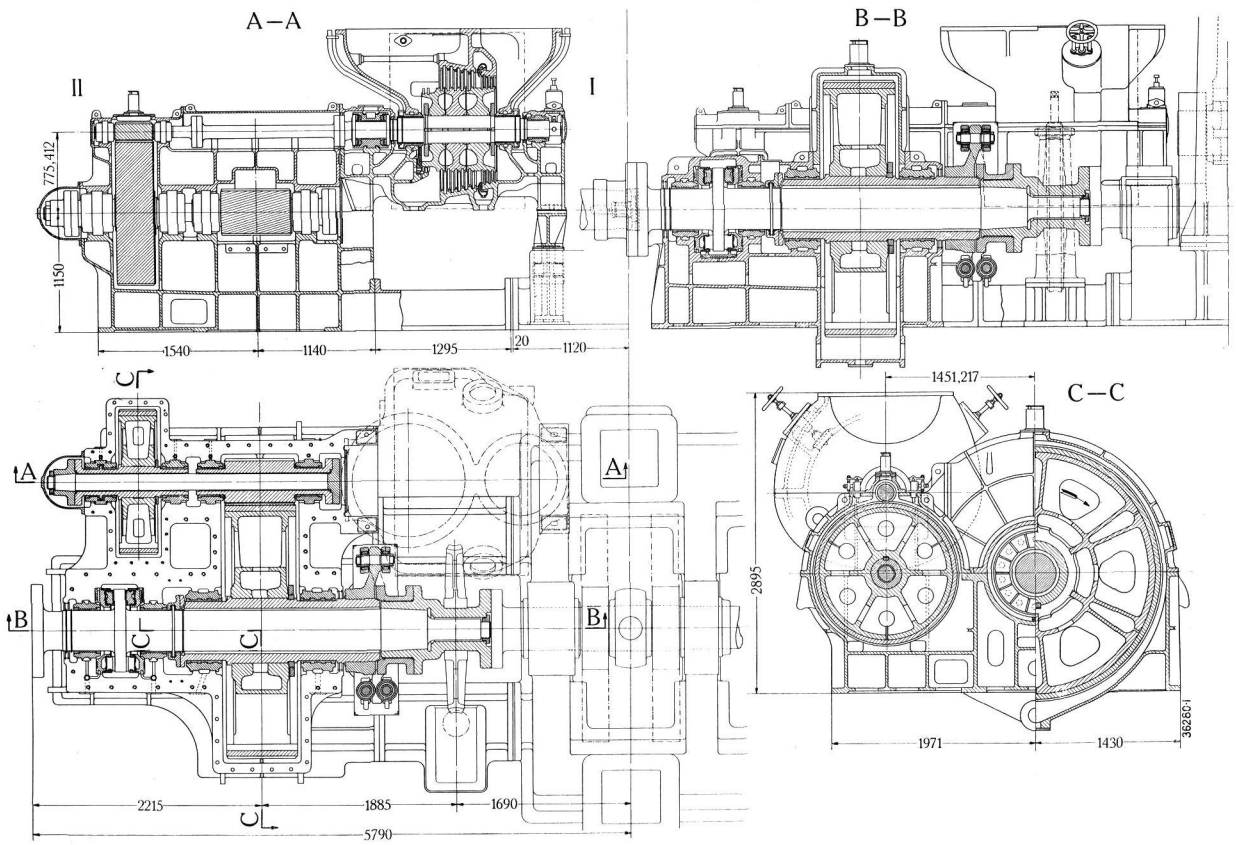


Fig. 2. — Exhaust steam-turbine of 1500 H. P., connected to a 4000 H. P. steam-engine. The turbine is coupled to the main shaft through gearing and a special Brown Boveri spring coupling.

I. Front. II. Back.

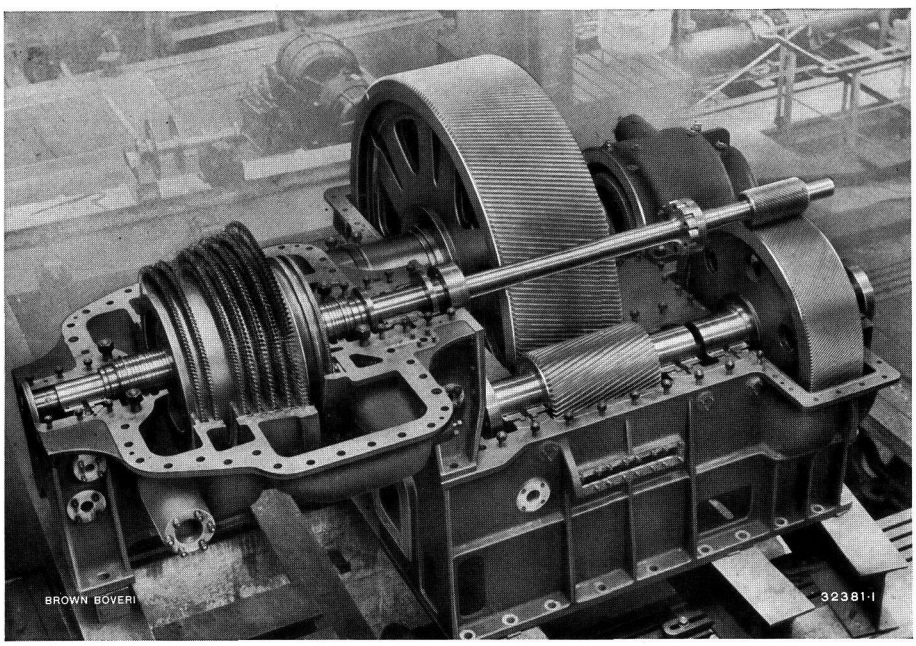


Fig. 3. — View, showing the exhaust steam turbine illustrated in fig. 2.

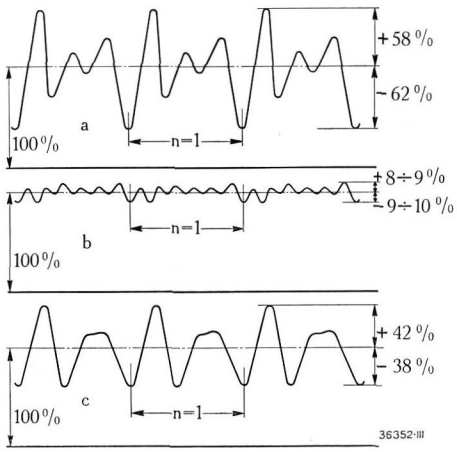


Fig. 4. — Crank-effort curves, measured at the propeller-shaft.

a. 2800 S. H. P. steam-engine of the "Kap Norte" without exhaust steam-turbine.

b. 4000 S.H.P. steam-engine of the "Ammon" with Brown Boveri exhaust steam-turbine.

Note the reduction in the torque fluctuations in case b, allowing the additional power of 1500 H. P. to be transmitted without strengthening the shaft.

c. The same curve for the steam-ship "Kap Norte", illustrated under a, but with Bauer-Wach turbine with hydraulic coupling.

and 8, which, however, refer to a pure turbine installation. In this example, one of seven cylindrical boilers, each having an output of 7·5 tons of steam per hour at 16 kg/cm² gauge and 300° C has been replaced by a Velox boiler for an output of 35 t/h at 50 kg/cm² abs, 450° C. The additional steam enables the power of the ship's engines to be doubled, i. e., to be increased from 10,000 to 20,000 H. P., although the steam quantity has been increased only by 80 %. The extra 20 % is provided by the higher specific output of the 50 kg/cm² 450° C steam, and by the higher efficiency of the Velox which is more than 15 % greater than that of the existing cylindrical boilers.

As can be seen from Fig. 8, the additional power is transmitted as follows: The Velox boiler supplies steam to a high-pressure turbine cylinder which delivers power through a new single reduction gear to the ships's shaft. An additional low-pressure turbine cylinder also delivers power to the shaft through the same gear. This low-pressure cylinder absorbs only part of the extra steam supplied by the Velox boiler, the remainder going to the existing low-pressure turbine, and thence to the condenser. The 100 % increase in power thus obtained enables the speed of the ship to be increased from 15·5 to about 20 knots. The conversion of this East Asia steamer has been proved successful by a number of trips, after a few troubles, due to obstruction of the combustion chamber by soluble salts present in the fuel, had been overcome.

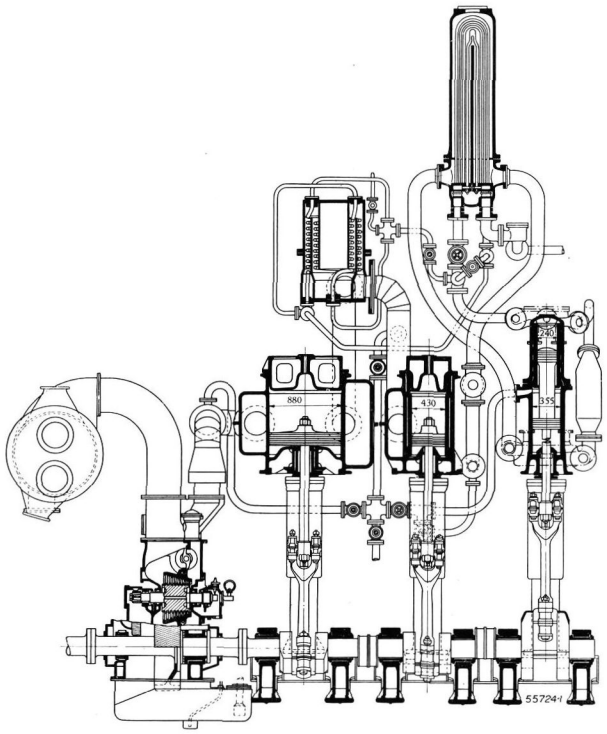


Fig. 5. — Quadruple-expansion engine of 1000 I. H. P., according to Prof. G. Bauer.

With this arrangement steam at 46 kg/cm² gauge at a temperature of 550° C is suggested, which is made bearable for the steam-engine by using the superheat of the live steam before entering the high-pressure cylinder for reheating the steam between the second and third expansion stages.

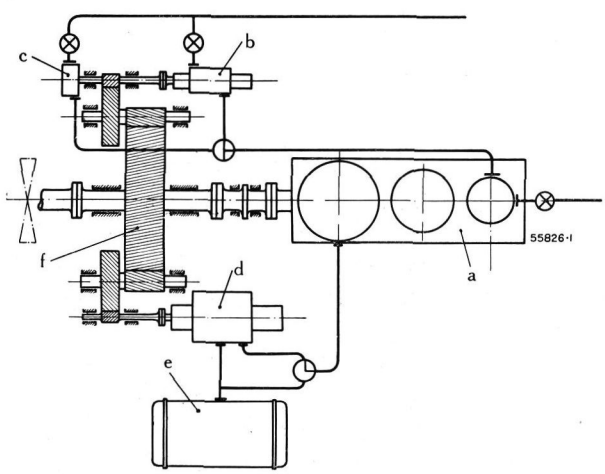


Fig. 6. — 3000 H. P. steam-engine installation with high-pressure primary turbine and low-pressure exhaust turbine.

This arrangement allows high-pressure, high-temperature steam to be used. The steam engine receives the steam at 17 kg/cm² gauge, 320° C.

a. Steam engine.
b. Ahead primary turbine.
c. Astern primary turbine.
d. Exhaust turbine.
e. Condenser.
f. Double reduction gear.

Returning to the steam engine, mention may be made of a new development, which aims perhaps more at a decrease in weight than at an improvement of the efficiency. This refers to designs, which following the

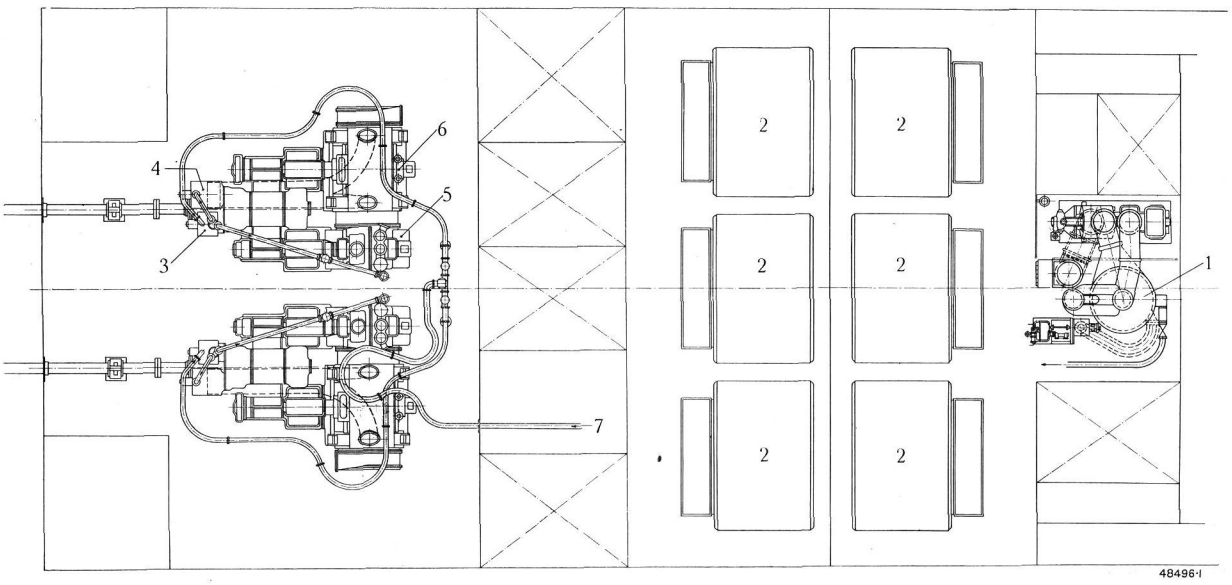


Fig. 7. — Power plant installation of the "Athos II".

One of the seven cylindrical boilers, each having an output of 7·5 t/h at 16 kg/cm² gauge, 300° C, equivalent to a total output of 10,000 H. P., was replaced by a Velox boiler having an output of 35 t/h at 50 kg/cm² gauge, 450° C, enabling the power to be increased from 10,000 to 20,000 H.P.

1. Velox steam-generator.
2. Cylindrical boiler.
3. Additional H. P. turbine.
4. Additional L. P. turbine.
5. Existing H. P. turbine.
6. Additional L. P. turbine.
7. Steam-pipe from Velox boiler.

example of internal combustion engines (Diesel and petrol engines) are characterized by a large number of cylinders in parallel, in order to achieve high speeds and hence low weights and small space requirements. Examples of such designs are afforded by the high-pressure locomotive of Buchli[1] and the Lenz radial engine[2].

It is clear that the advantages obtained by increasing the speed of steam engines cannot be compared with those obtained in the case of combustion engines, because the boiler and the condenser with their auxiliaries are not affected. If, however, the advantages which are offered in this respect by the Velox boiler, are combined with those of this new type of engine, quite interesting installations may result, and this brings us to

THE BOILER.

The boiler has a decisive influence on the space requirements, a considerable influence on the price, and an appreciable effect on the efficiency of the installation.

Boiler development is closely connected with the advances in pressure and temperature in turbine design, so that it is advisable to consider these first. Fig. 9 shows the way in which the pressures have risen since the time that Brown Boveri first took up the manufacture of turbines in the year 1901 up to the present date. The diagram contains two curves, one referring to the maximum pressure occurring during the course of the year under consideration, the other referring to the average pressure of the turbines built during that year by Brown Boveri and their concessionaries. It can be seen that there is a certain continuity in the progress of both pressures, which is of course more marked in the case of the average pressure. On one occasion this continuity was sharply interrupted when a turbine built for a pressure of 200 kg/cm² caused a high peak in the curve of maximum pressures. This was a turbine intended for driving the feed pump of a Benson boiler at Langerbrugge in Belgium in the year 1929 that is, at a time when the Benson boiler really did work according to the original Benson idea, namely, at the critical pressure. If this peak is left out of account we have the following picture: From the beginning of turbine manufacture up to the year 1921, the average pressure increases along an almost straight line from 10 to 13 kg/cm². Only then does the rise begin to take place more rapidly along a line which is again approximately straight until the year 1928, when a pressure of 25 kg/cm² is attained, after which follows a rather irregular increase up to the pressure of 54 kg/cm² at the beginning of the war. Since then, the average absolute pressure has fallen by about 10 kg/cm² to 44 kg/cm²; this being in all probability due to difficulties in obtaining the required materials. The maximum pressure has, apart from the 200 kg/cm² peak already mentioned, risen in the same interval to 135 kg/cm² abs.

[1] Buchli: Hochdruck-Lokomotive "Winterthur". Schweizerische Bauzeitung 2nd June, 1928.

[2] F. Clar: Schnellaufende Dampfmaschinen, ZVDI 1940, page 769.

Equally interesting is the curve of live steam temperatures (Fig. 10). Here also, both the average and the maximum yearly values have been drawn. The rate of rise of the average temperature is sensibly constant between 275 and 330 ° C during the years 1901 to 1923. The increase is rather less regular from then onward up to 440 ° C in the year 1940.

The maximum temperatures follow, as might be expected, a more irregular curve. Up to 1923, the maximum temperature did not exceed 375 ° C after which it rose rapidly to reach 500 ° C in the year 1930. At this moment, a certain drop took place due to difficulties experienced with superheater tubes, and it was not until the year 1935 that a maximum temperature of 500 ° C was again attained, at which value it has remained practically constant up to the present day.

The call for higher pressures and temperatures has always come from the turbine makers, and boiler manufacturers followed this demand but reluctantly, until Benson hit upon the idea of building a boiler for the critical pressure and of operating it at this pressure. A few years previously, the Swede Blomquist had attempted with his Atmos boiler to create a typical high-pressure boiler in which the evaporation portion consisted of rotating tubes. Because of these rotating tubes, however, this did not achieve much success in practice.

Since then, the principle of operating the boiler at critical pressure has been abandoned so that the Benson boiler has reverted to the usual form of high-pressure boiler, such as those of Doble and Sulzer. These drumless "all-tube type" boilers meet the requirements of high pressures because they possess no drums and need relatively little material for their manufacture. All of these boilers have forced water circulation.

The Velox boiler employs the forced circulation principle not only for the water but also for the air and combustion gases. It is, apart from a boiler built for the French navy by its chief engineer Monsieur Norguet, the only supercharged boiler, i. e., the only boiler where combustion takes place under pressure. In regard to efficiency, low weight, and small space requirements, it is unequalled by any other form of boiler and it enables full use to be made of the progress made in these directions by steam prime-movers. In spite of its small dimensions, it is fully 3—5 % better in efficiency than the best other types of boiler. Up to the present, over 100 Velox boilers have been built but only for oil or gas fuels. Should it, however, be possible to construct a Velox boiler for coal firing, as seems not unlikely, a revolution in the designs of power plants is likely to result, as the large boiler houses hitherto usual will no longer be necessary and the boilers will be erected next to the turbines (see Figs. 11 and 12), as has already been done in the case of oil-fired units.

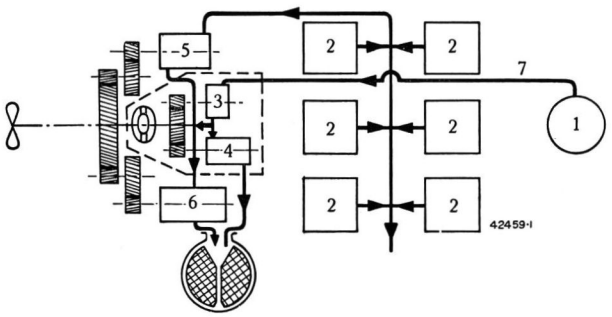


Fig. 8. — Diagrammatical arrangement of the installation shown in Fig. 7.

The extra output of a Velox boiler of 35 t/h output at 50 kg/cm² gauge, 450 ° C, is used in an additional high-pressure and in an additional low-pressure cylinder to develop power which is transmitted through the hydraulic coupling to the existing gear.

1. Velox steam generator.
2. Cylindrical boiler.
3. Additional H. P. turbine.
4. Additional L. P. turbine.
5. Existing H. P. turbine.
6. Additional L. P. turbine.
7. Steam-pipe from Velox boiler.

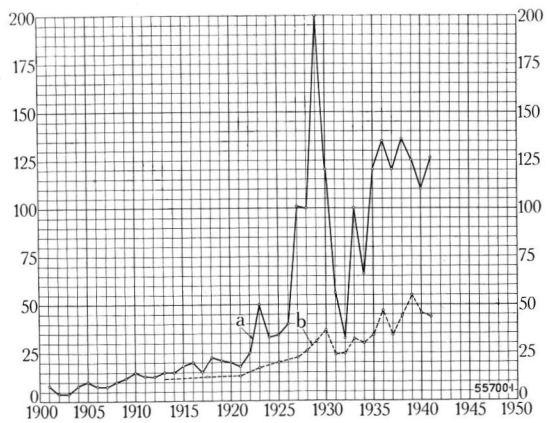


Fig. 9. — Live steam pressure of steam-turbines built by the Brown Boveri Concern from 1901 to 1941.

The curves of the maximum and average pressures show a moderate increase up to 1921 and from that moment a more rapid increase up to about 50 kg/cm² abs average pressure and 125 kg/cm² abs maximum pressure in the year 1940.

Live steam pressure: a. Maximum. b. Yearly average.

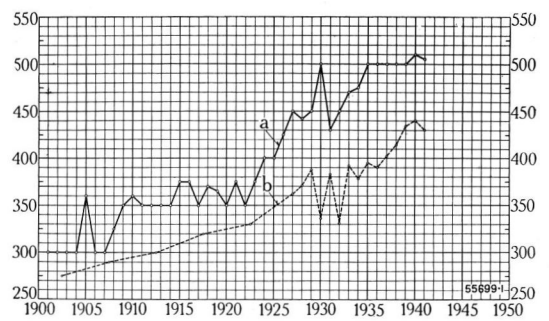


Fig. 10. — Live steam temperatures of the steam-turbines built by the Brown Boveri Concern from 1901 to 1941.

The curves show a steady increase in the average temperatures of the turbines supplied by the Brown Boveri Concern up to the year 1922 and a maximum temperature of 300 ° C up to 1907 and 320 ° C up to 1922. From 1922 onward, the temperature rose rapidly until the year 1941 with a few set-backs around 1930, to attain an average temperature of 440 ° C in 1940 and a maximum temperature in the same year of 510 ° C.

Live steam temperature: a. Maximum. b. Yearly average.

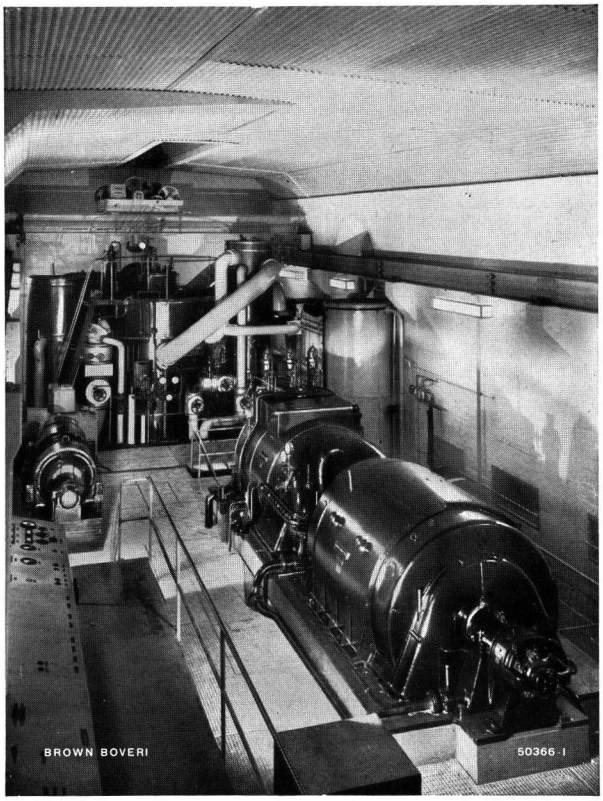


Fig. 11. — Velox steam-generator in the 9000 kW bomb-proof power station of the city of Berne.

A Velox boiler, installed in the same room as the steam turbine, enables the full-load output of the plant to be attained within ten minutes of starting from cold.

THE STEAM TURBINE.

The reason why the steam turbine has displaced the steam engine so quickly and to such a large extent, is primarily because it enables a far better use to be made of high vacua; the high-pressure part was in regard to efficiency originally far behind the steam engine, especially for the small powers formerly coming into consideration. This was also the reason for the extraordinarily slow rise in pressures and temperatures visible in the curves of Figs. 9 and 10.

As the increase in efficiency of the steam turbine proved a difficult matter and could be achieved only after long and laborious improvements of the nozzles, blading, etc., attention was turned more and more towards the possibilities of improvement offered by increasing the heat drop. As already mentioned, because of the difficulties experienced with available materials it has so far only in exceptional cases been possible to go above a temperature of 500 ° C, and then only temporarily. On the other hand, the pressures have risen considerably, especially in back-pressure installations where the exhaust steam is used either in existing turbines or for process purposes in chemical and other industrial plants. In the case of the latter, in particular, it is often worth while to go up to very high pressures in order to be able to furnish the energy requirements of the plant so to speak in the form of by-product power of the steam supply. In such plants it is possible to attain, apart from the boiler losses, a practically 100 % utilization of the heat of the steam because all the losses in the turbine go to raising the temperature, and hence the heat content of the process steam. This fact has led to a considerable increase in the number of back-pressure turbine installations, compared with condensing turbines and it is to be expected that this tendency will be even more marked after the war due to the higher price of coal. Plans are under consideration for combining electric supply stations with large district heating plants in order to obtain the heating as a cheap by-product where energy generation is looked upon as the prime object and cheap by-product energy where heating is the main object.

Mention may here be made of a special form of heat pump suggested by Prof. B. Bauer of Zurich.[1]

According to this suggestion a back-pressure turbine drives a heat pump which serves for the first heating stage, the heat of the exhaust steam from the turbine

[1] Neue Zürcher Zeitung, Dec. 10th, 1941.

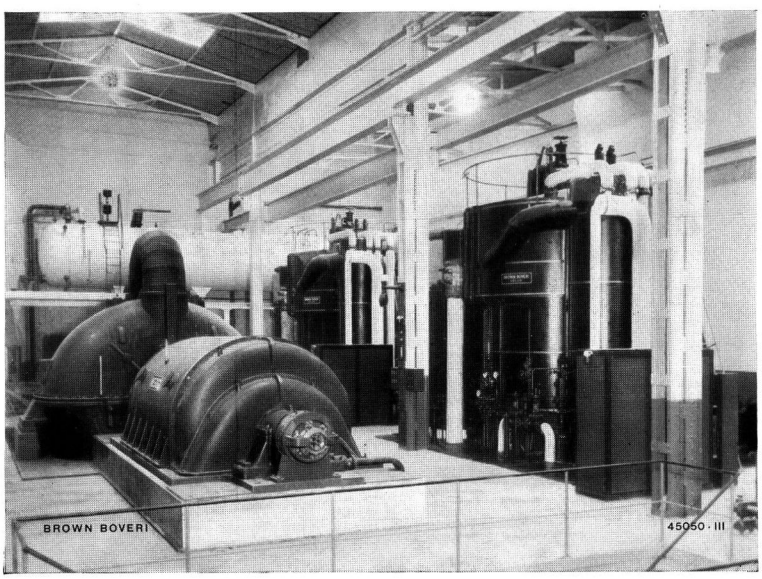


Fig. 12. — Two Velox boilers for a steam quantity of 75 t/h are installed in the same room as the turbine in the electric generating plant of the city of Oslo.

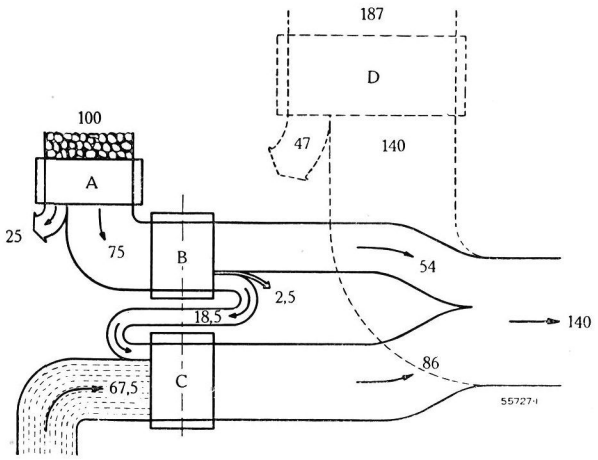


Fig. 13. — Remote heating plant with back-pressure turbine.

a. High-pressure boiler. c. Heat pump.
b. Steam-turbine. d. Low-pressure boiler.

According to a suggestion of Prof. B. Bauer of Zurich, the back-pressure turbine drives a heat pump and thereby increases the amount of heat, supplied to 2½ times that which would otherwise be available.

effecting the remaining temperature increase to the desired value which depends on the atmospheric temperature. In this manner it is possible after deducting the boiler and negligibly small bearing losses of the turbine to utilize up to 80 % of the heat in the fuel. The equivalent of the heat drop over the back-pressure turbine is transmitted in the form of power to the compressor and appears in the form of compression heat energy in the working medium of the heat pump from whence it is transmitted to the heating water. The heat energy of the steam not given up in the turbine is yielded directly to the heating water in the condenser in which both the latent heat of condensation as well as the losses in the turbine blading are recovered. To this is added some four to five times the heat equivalent of the power output of the turbine in the form of heat energy extracted from the surroundings by the heat pumps; that is to say the total heat obtained is some 130 to 165 % of the heat contained in fuel fired in the boiler.

How can now the further development of the power turbine be visualized? A further increase in steam pressure above the value of 140 kg/cm² at present common does not (as can be seen from Fig. 14) bring very much gain with a steam temperature of 500° C. On the other hand, as seen from Fig. 15, a further rise of the steam temperature at the same pressure shows a promising improvement, and points the way to

THE HOT-STEAM TURBINE.

Experience with gas turbines shows that such machines can safely be operated at temperatures of 600° C

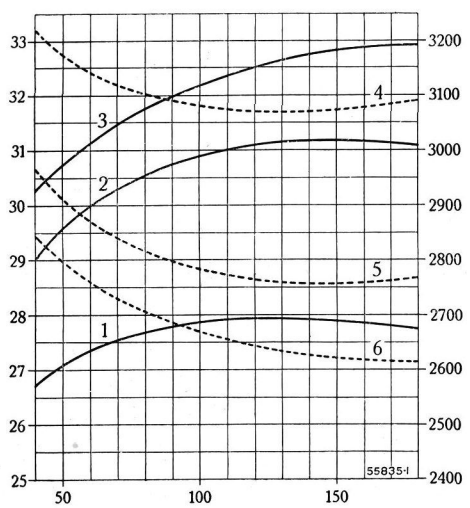


Fig. 14. — Effect of the live-steam pressure on the thermal efficiency and heat consumption of a 50,000 kW turbo-set with a live-steam temperature of 500° C.

——— Thermal efficiency coal to generator terminals.*
---------- Heat consumption coal to generator terminals.*

* Including power absorbed by feed pump, condenser auxiliaries and basing on a boiler efficiency of 86 %. Reheating up to the live-steam temperature.

1 and 4. Plant without feed heating and without reheating.
2 and 5. Plant with feed heating and without reheating.
3 and 6. Plant with feed heating and with reheating.

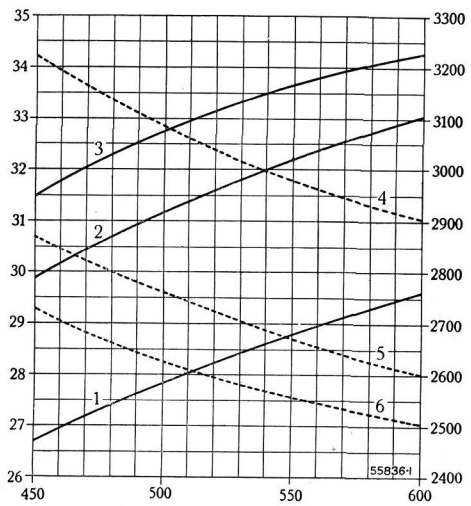


Fig. 15. — Effect of the live-steam temperature at a pressure of 140 kg/cm² abs on the thermal efficiency and heat consumption of a 50,000 kW set.

——— Thermal efficiency coal to generator terminals.*
---------- Heat consumption coal to generator terminals.*

* Including power absorbed by feed pump, condenser auxiliaries and basing on a boiler efficiency of 86 %. Reheating up to the live-steam temperature.

1 and 4. Plant without feed heating and without reheating.
2 and 5. Plant with feed heating and without reheating.
3 and 6. Plant with feed heating and with reheating.

and suggests the advisability of investigating whether steam turbines could also be operated at such temperatures and what efficiencies might be expected to result therefrom. For this purpose three alternative propositions were studied:—

(1) High superheat to 600° C at a high pressure of 140 kg/cm² abs with straight expansion to a vacuum of 0·04 kg/cm² abs.

(2) Expansion from 140 kg/cm² abs at a moderate superheat of 450° C down to 35 kg/cm² abs and 273° C, then reheating to 600° C with pressure loss of 5 kg/cm² to 30 kg/cm² abs and expansion to a vacuum of 0·04 kg/cm² abs with the object of not cumulating the difficulties due to high pressure with those of high temperature.

(3) High pressure 140 kg/cm² abs at a high superheat of 600° C; expansion down to 35 kg/cm² abs at 404° C; reheating to 600° C with a pressure loss of 5 kg/cm² abs down to 30 kg/cm² abs and expansion down to a vacuum of 0·04 kg/cm² abs.

As can be seen from the Mollier diagram (Fig. 16) and from table I, the resulting efficiencies, referred to the heat content of the live steam at the conditions mentioned, to power developed at the turbine coupling allowing for all turbine losses (gland loss + blading loss + radiation + bearing loss) are

Alternative 1:

Efficiency 41·25%; heat consumption 2085 kcal/kWh.

Alternative 2:

Efficiency 41·52%; heat consumption 2070 kcal/kWh.

Alternative 3:

Efficiency 43·25%; heat consumption 1987 kcal/kWh.

In all cases, a final feed temperature of 218° C obtained by regenerative feed heating with extracted steam was assumed.

If we take into account the losses of the auxiliaries and of the boiler as well as the reheater, we obtain the figures given in Table II, according to which we obtain for alternative 3, with the usual type of boiler and reheater, an efficiency of 35% referred to the heat contained in the fuel and the energy at the generator terminals, corresponding to a heat consumption of 2457 kcal/kWh. In the case of an oil-fired Velox the efficiency is increased to 36·6% whilst the heat consumption falls to 2350 kcal/kWh.

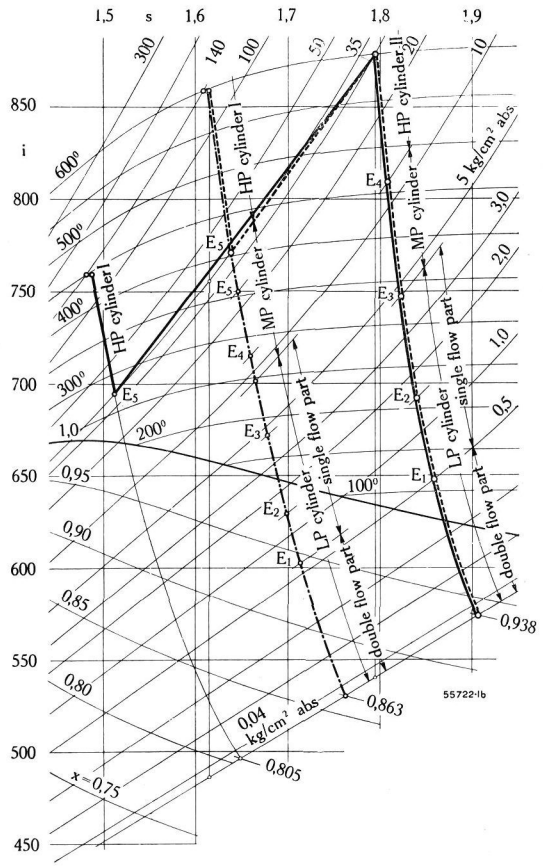


Fig. 16. — Expansion lines of a 40/50,000 kW turbo-set at 40,000 kW output.

—·—·— Alternative 1 of table I.
——— Alternative 2 of table I.
---------- Alternative 3 of table I.

1. Expansion from 140 kg/cm² abs, 600° C, without reheating down to condenser vacuum.
2. Expansion from 140 kg/cm² abs, 450° C to condenser vacuum with reheating at 35 kg/cm² abs from 273 to 600° C.
3. Expansion from 140 kg/cm² abs, 600° C to condenser vacuum with reheating at 35 kg/cm² abs from 404 to 600° C.

These figures are, so far, purely theoretical, although they are based on the readily estimable losses of the boiler, the reheater and the turbine. It remains for a practical installation to prove these results.

We must first decide which of the three alternatives we will select. According to the figures of Table II alternative 3 is the most advantageous as it results in the highest efficiency and the lowest heat consumption. One might, however, be tempted to choose alternative 2 since the difficulties seem to be less owing to the fact that the high temperature (600° C) is encountered only in the reheater, that is where the pressure is no longer high. Investigation shows, however, that it is easier to build a high temperature superheater for a high pressure than for a low one, because the heat transfer is better and

TABLE I.

Expansion Conditions for a 40/50,000 kW Turbo-set at 40,000 kW.

Alternative	1	2	3	
Live steam	140/600	140/450	140/600	kg/cm² abs/° C
Heat content	859·0	759·75	859·0	kcal/kg
Reheating at	—	35/30	35/30	kg/cm² abs
from/to t_{reheat}	—	273/600	404/600	° C
Circulating water	15	15	15	° C
Corresponding vacuum	0·04	0·04	0·04	kg/cm² abs
Total heat drop	373·5	417·5	442·5	kcal/kg
Turbine	Three-	Four-	Four-	cylinder
Speeds: H. P.-cyl. I and H. P.-cyl. II	9000/—	9000/9000	9000/9000	r. p. m.
M. P.- and L. P.-cyl.	3000/3000	3000/3000	3000/3000	r. p. m.
Heat consumption at coupling	2085	2070	1987	kcal/kWh
η_{th} referred to live steam conditions and coupling output	41·25	41·52	43·25	%
Heat supplied: to the live steam	636·0	536·75	636·0	kcal/kg
in the reheater	—	165	98·7	kcal/kg
Total	636·0	701·75	734·7	kcal/kg
Steam consumption referred to output at coupling	3·28	2·95	2·705	kg/kWh
Total	∼135	∼121	∼110	t/h

Feed-water heating in five stages to 218° C or 223 kcal/kg.

TABLE II.

40/50,000-kW Turbo-set.

Summaries of the Steam- and Heat-consumption Figures and of the Thermal Efficiencies for Operation at 40,000 kW.

Boiler and reheater	Normal			Velox		
Alternative	1	2	3	1	2	3
Live steam at turbine kg/cm² abs/° C	140/600	140/450	140/600	140/600	140/450	140/600
Feed-water heating to ° C	218	218	218	100	100	100
Reheating	no	yes	yes	no	yes	yes
from kg/cm² abs/° C		35/273	35/404		35/273	35/404
to kg/cm² abs/° C		30/600	30/600		30/600	30/600
Steam consumption referred to terminal output including feed heating kg/kWh	3·37	3·03	2·78	2·96	2·62	2·45
Thermal efficiency with feed heating (referred to conditions of turbine inlet and output at generator terminals) %	40·2	40·5	42·2	38·25	38·9	40·5
Auxiliaries and losses between boiler and turbine %	3·5	3·8	3·5	2·7	2·9	2·7
Thermal efficiency including feed-heating (referred to generator terminals), outlet from boiler and from reheater, including losses of the auxiliaries and between boiler, reheater and turbine %	38·8	39·0	40·7	37·2	37·8	39·4
Efficiency of the boiler and the reheater %	86	86	86	93	93	93
Thermal efficiency coal to terminals including boiler and reheater efficiency %	33·4	33·5	35·0	34·6	25·1	36·6
Heat consumption coal to terminals kcal/kWh	2575	2568	2457	2485	2450	2350

Steam- and heat-consumption figures as well as thermal efficiencies of a 40/50,000 kW turbo-set at an output of 40,000 kW.

Note particularly the last two lines giving the thermal efficiency and the heat consumptions, referred to the heat-content of the coal and the output at the generator terminals, including all losses of the boiler, turbine, generator, condensing plant, and auxiliaries.

hence the wall temperature lower. Moreover, with high-pressure steam, the heating surfaces and the tube diameters are smaller for the same pressure drop, so that particularly in the case of Velox superheaters operating with high gas velocities, the total weight is less and the erection and accommodation of the tubes are easier for a higher pressure than for a lower one. We, therefore, decide in favour of alternative 3, and will in the following consider only this alternative.

The practical realization of this solution is conceived as follows (Fig. 17):

A normal type of boiler furnishes steam at the conditions usually met in high-pressure installations, for instance, 150 kg/cm² abs, 450° C. The temperature is deliberately chosen on the low side; it could be even still lower, for example 400° C; to make absolutely sure of avoiding any superheater difficulty in the boiler and to permit the use of ordinary steel for the superheater tubes. This boiler may then be equipped with a liberally dimensioned air preheater such as is usual when the feed water is raised to a high temperature, in this case 218° C, by extracted steam.

Both the additional high superheating and the high temperature reheating are effected in a separately fired special superheater B operating on the Velox principle. Although the Velox principle may be assumed to be generally known, it may briefly be mentioned that in this Velox superheater, as in the Velox boiler, combustion takes place under pressure in the combustion

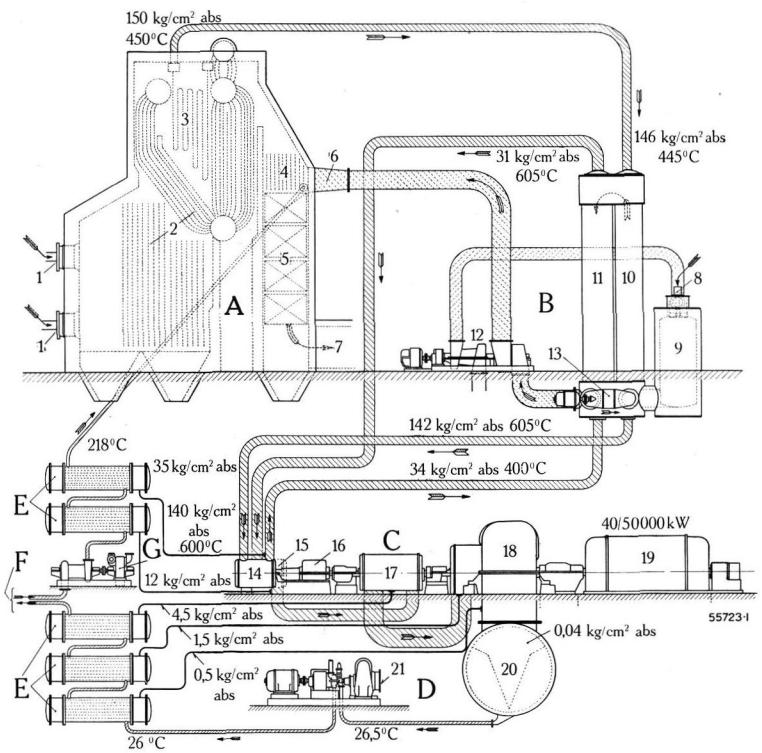


Fig. 17. — Diagrammatic arrangement of a 40/50,000 kW steam turbine plant, with conventional boiler A and a special superheater B on the Velox principle for raising the temperature of and reheating the steam.

The turbine C has four cylinders, the power of the first two being transmitted to the shaft through a reduction gear.

A. Steam boiler.

1. Pulverized fuel burner.
2. Evaporator heating surfaces.
3. Superheater.
4. Economizer.
5. Air heater.
6. Exhaust gas duct from superheater B.
7. Exhaust gas duct to chimney.

B. Additional and reheating superheater.

8. Fuel inlet.
9. Combustion chamber.
10. Additional superheater.
11. Reheater.
12. Charging set.
13. Exhaust-gas recirculating blower.

C. Steam turbine.

14. High-pressure cylinder I.
15. High-pressure cylinder II.
16. Gear.
17. Intermediate-pressure cylinder.
18. Low-pressure cylinder.
19. Generator.

D. Condenser.

20. Condenser.
21. Condenser auxiliaries.

E. Extracted-steam feed-water heater.

F. Connection to feed-water equalizing tank.

G. Feed pump.

chamber 9 and the combustion gases flow at high velocity through the superheater 10 and the reheater 11 so that high rates of heat transfer are obtained, the necessary air pressure being produced by the axial-flow compressor of the charging set 12, driven by a gas-turbine which is actuated by the combustion gases. The latter enter the turbine at a still high temperature and at the charging pressure diminished by the resistance drop in the tubes. The exhaust gases from the turbine may be conducted through the pipe 6 to a suitable point in the boiler. Alternatively, it may be more convenient to use the gases which leave the turbine at about 380° C, for heating the feed water in a pre-heater, which takes the place of the fifth heater E fed with extraction steam from the turbine. In order to limit the maximum gas temperature in the superheater and to ensure against burning of the superheater tubes, it is proposed to recirculate the combustion gases by means of an axial-flow blower, which has only to overcome the resistance drop in the superheater and reheater.

This arrangement has the following advantages:

(a) By regulating the burner 8 of the Velox superheater the superheat can be regulated independently of the load and of the combustion in the boiler A, so that it remains practically constant. This is exceedingly important since variations of the order of 50° C, such as may occur in the superheat of a boiler of usual design operating at 600° C, cannot be tolerated.

(b) Due to the employment of the Velox principle, heating surfaces are considerably reduced; they amount for the same pressure drop on the steam side, to only $^1/_4$—$^1/_5$ of the size necessary with firing at atmospheric pressure. The quantity of high temperature steel required for the tubes and hence the cost is reduced correspondingly.

(c) The fact that the steam enters the external superheater at a relatively high temperature results in an only partial utilization of the heat in the combustion gases, so that they leave the superheater at a relatively high temperature. With the Velox superheater a considerable part of the remaining heat, corresponding to a temperature drop of 140 to 160° C, is absorbed in the gas turbine and returned to the combustion chamber in the form of compression energy supplied to the combustion air.

(d) The heat still remaining can be used, for instance, for heating the feed water, this heater taking the place of the first extracted steam heater or being connected in parallel with the first two extraction heaters. This is all the more possible, because due to the use of such a high reheat temperature it is hardly necessary to extract any water from the low-pressure stages of the turbine. This heater also makes use of the Velox principle requiring for the same heat transfer a heating surface approximately ten times smaller than would be necessary in the usual form of economizer.

(e) By recirculating the gases, i. e., by reducing the temperature of the fresh combustion gases by mixing them with the recirculated gases, and by making the best possible use of the exhaust heat, efficiencies can be attained which like those of the Velox boiler are of the order of 90 % and over.

(f) Because of the forced circulation of the combustion gases it is possible to operate with large pressure drops on the gas side, thus preventing any possibility of excessive temperatures in certain tubes or tube banks due to unequal external heating; the same object is achieved on the steam side by allowing an adequate pressure drop for the steam flowing through the tubes.

(g) The separate superheater can be placed near to the turbine so that the high-pressure and high-temperature pipe connections can be kept very short.

(h) The separate firing of the Velox superheater enables the temperature to be increased slowly, thus avoiding difficulties during starting.

(i) In case of trouble with the separate superheater, the turbine may be operated with steam straight from the boiler at a correspondingly reduced load.

In Fig. 17, the steam which leaves the boiler A at a pressure of 150 kg/cm² abs and a temperature of 450° C goes to the separate superheater 10, where its temperature is raised to 605° C, so that it enters the first cylinder of the four-cylinder turbine C at a pressure of 140 kg/cm² abs, and a temperature of 600° C. After expansion in the cylinder 14 it flows to the reheater 11 which it enters at a pressure of 34 kg/cm² abs and a temperature of 400° C and leaves at 31 kg/cm² abs, 605° C to enter the second cylinder 15 of the turbine (placed behind the cylinder 14) at 30 kg/cm² abs, 600° C. From there the steam flows to the cylinder 17 at 4·5 kg/cm² abs, 350° C which it leaves at 1·5 kg/cm² abs, 200° C to enter the low-pressure cylinder 18 where it expands down to a vacuum of 0·04 kg/cm² abs in the condenser 20. The wetness

at the end of expansion is only 9·38 %, a figure which as experience shows, does not lead to any trouble due to erosion of the low pressure blading.

Turning now to the turbine which is illustrated in Fig. 18, the first high-pressure cylinder (high-pressure, high-temperature cylinder) is of single-flow design and runs at 9000 r.p.m., the second high-pressure cylinder

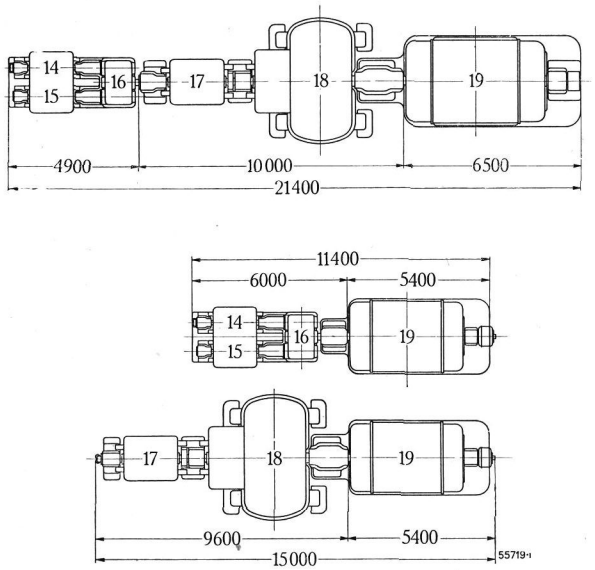


Fig. 18. — Dimensions of a 40/50,000 kW hot-steam turbo-set, alternative single and two-shaft arrangments.

Two high-pressure cylinders are connected in front of a two-cylinder turbine. These units run at 9000 r. p. m. and transmit their power through a reduction gear.

is of the double-ended type so that the blade length will be of the same size as that of the first high-pressure cylinder, thus enabling it to be operated with the same speed. It may seem paradoxical to operate these two high-temperature cylinders, one which is also subjected to the highest pressure at this high speed. Closer consideration shows, however, that this is justified in every respect as it enables the dimensions of the cylinders proper and hence the thickness of the walls as well as the length of blades to be kept small. The power developed by these two cylinders is transmitted through a reduction gear to the shaft of the intermediate-pressure cylinder which like the low-pressure cylinder and the generator runs at a speed of 3000 r. p. m.

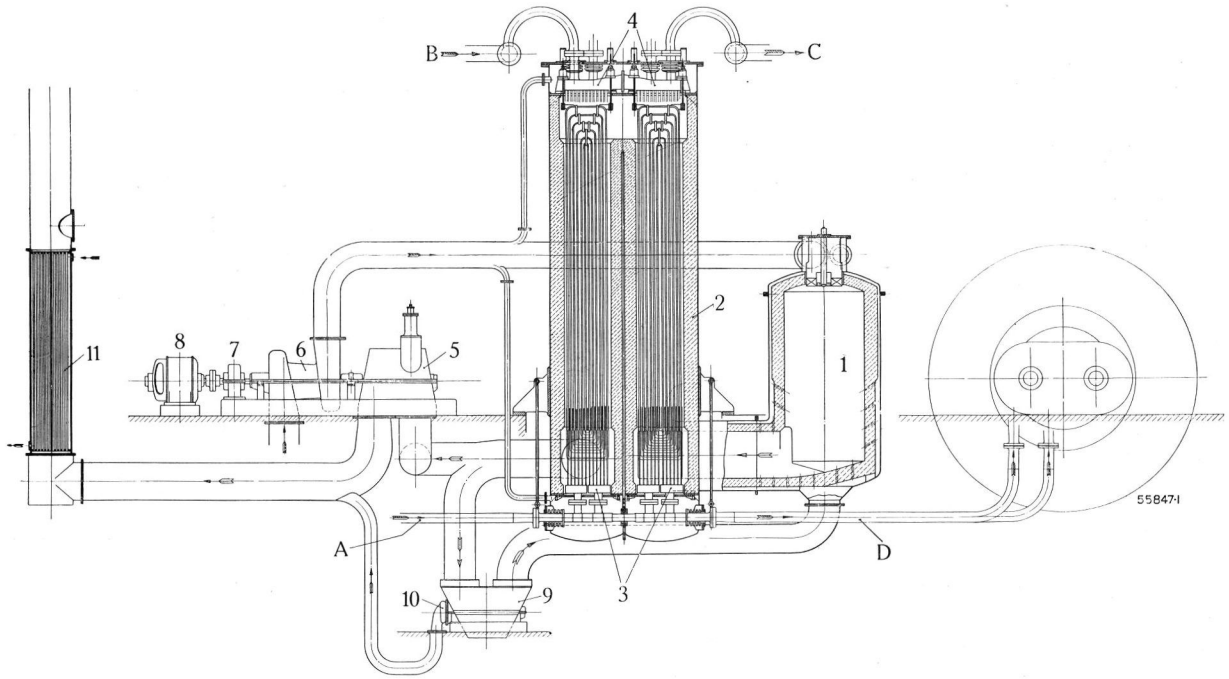
In working out an example of the design of such a machine the output was intentionally chosen at the high value of 40,000—50,000 kW because smaller units would be easier to build. This study has shown that the realization of such a unit with the means at present known, that is, with the high-temperature steels such as were available before the war, and with existing construction elements, is readily possible.

One point should be mentioned here, which is often raised against the use of high steam temperatures. This concerns the decomposition of steam into hydrogen and oxygen and the danger of corrosion of the superheater tubes by nascent hydrogen. Such a decomposition does not take place at the temperatures which we propose because in spite of the high heat transfer in the Velox superheater the internal temperature of the tube wall, and hence of the boundary layer is only about 50° C higher than the steam temperatures, i.e., only about 650° C, whereas the decomposition of steam does not begin until about 1200 to 1400° C.

On the other hand, the chemical reaction $3\,Fe + 4\,H_2O = Fe_3O_4 + 4\,H_2$ is definitely a function of the temperature. Already at the temperatures of 500° C common at the present time, cases are known where serious troubles have occurred due to this cause. Thus, for instance, it is known to the author, that the American Port Washington power station which has since become renowned in the U. S. A. as having the smallest consumption of kcal per kilowatt-hour could not be put into continuous service for a long while after starting up for this very reason. By slowly raising the pressure and temperature of the boilers with a corresponding slow increase in the output of the plant it was then attempted to form a permanent oxide coating on the tubes in order to render the surface of the material immune. This measure was entirely successful and from then onward no further difficulties were experienced.

It is extremely important from the point of view of creep strength that the steels which are used for the separate superheater should have a high chromium content, so as to be highly resistant to corrosion, since this chromium content favours the formation of a protective coating consisting of double oxide of chromium and iron. Tests made in this direction by Hatfield in England, show that a steel of this type used for 700° C has a thirty-seven times greater resistance to corrosion than a tube of ordinary steel. According to Hatfield the time required to reduce the thickness of the walls of a chromium-nickel-tungsten tube, by 1 mm would be about twenty-two years. By rendering the internal surface immune this corrosion could be further reduced to a fraction if not eliminated entirely.

In regard to inter-crystalline corrosion, there is little to be feared with such a large chromium content when the amount of carbon is small.



Fig, 19. — Velox live-steam superheater and reheater for a 25,000 kW hot-steam turbine.

A. From steam boiler.
B. To the reheater.
C. From the reheater.
D. To the turbine.
1. Combustion chamber.
2. Superheater casing.
3. Additional superheater.
4. Reheater.
5. Gas-turbine.
6. Air-compressor.
7. Gear.
8. Starting and regulating motor.
9. Exhaust-gas recirculating blower.
10. Gas-turbine.
11. Exhaust-gas heat recuperator.

A very interesting "hot-steam" turbine plant of this type is about to be built in two stages with a continuous rating of 25,000 kW.

The pressure both for the first and final stage will be 150 kg/cm^2 abs at the boiler, 140 kg/cm^2 at the turbine. The temperature, on the other hand, will be at first 500° C after the superheater, and 480° C after the reheater, to enable the superheater to be built with standard tubes. Later, when suitable material is available for the tubes, it will be raised to 600° C both after the superheater and after the reheater. Only then, will the high-temperature high-pressure cylinder be added to the turbine. The additional power then available, will be about 9 %. The extra cost of the final installation compared with standard high-pressure plant for 500° C with superheater built into the boiler will be about 30 %, and will be recovered in approximately two years by the saving in coal achieved. A detailed drawing of the separate superheater of this plant is shown in Fig. 19.

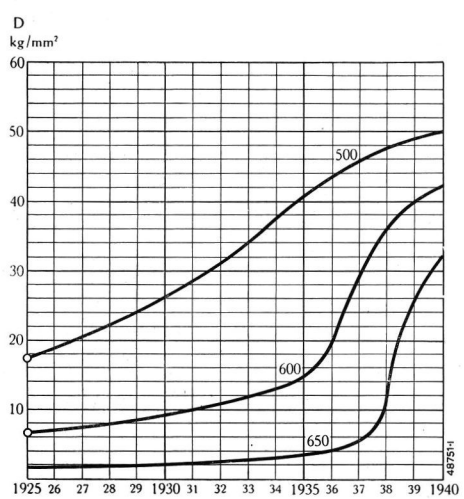


Fig. 20. — Austenitic steels.

Improvement of the creep strength at 500, 600 and 650° C during the years 1925 to 1940.

D. Measured max. values of the creep strength.

As has been seen, the development of steam power installations will depend mainly on a proper use of existing and of improved or newly invented high temperature steels. The considerable advances in this direction which were made in the years preceding the present war are illustrated in Fig. 20.

It has been shown how such steels can be appropriately applied. It would be a worthy task for research institutions and manufacturers to collaborate in this advance to the profit and advantage of our highly developed thermal engine industry.

(MS 957) *Dr. Ad. Meyer. (Hv.)*

SOME APPLICATIONS OF THE HEAT PUMP AS A HEATING MACHINE.

Decimal Index 621.577

This article completes the theoretical explanations given in the May/June 1943 number of this journal by a study of the various applications of the heat pump for heating and by a description of installations carried out. The conditions under which a heat pump can come into consideration with advantage are also examined.

With the exception of the steam-turbine-driven heat pump and evaporating plants, the heating conditions assumed apply primarily to Switzerland, where great efforts have been made and considerable ingenuity expended to heat without imported fuel which is not only expensive, but also, at the present time, exceedingly hard to obtain. The same conditions and conclusions apply, however, to many regions where hydro-electric energy is abundant and fuel comparatively expensive.

I. WHEN CAN A HEAT PUMP COME INTO CONSIDERATION FOR HEATING?

THE general answer to this question is simply: whenever heat under moderate temperatures is required — not exceeding the boiling point of water under atmospheric pressure, for instance — in form of hot water or air for heating, washing or drying in industrial plants, as well as for heating rooms and air conditioning. For all cases of this description, the advantages of the heat pump should be carefully examined. Somewhat different considerations apply for evaporating plants with thermo-compressors, which are reviewed in chapter VII.

With regard to the utilization of available energy, the practice of lowering the value of high-grade energy must be condemned. Common examples are afforded by burning fuel in a boiler at a high temperature, which has to be subsequently reduced by resistance to the transmission of heat or in pipelines and by consuming electricity which represents energy in its highest form, in heating resistances. In such cases, high-grade energy should be consumed in order to raise the value, by increasing the temperature to the desired extent, either of redundant heat or of heat taken from the surroundings[1]. This stepping-up process is carried out by a heat pump, against an expenditure of high-grade energy which amounts to only a fraction of that necessary to produce the same amount of heat directly, namely $^1/_3$ to $^1/_6$ for heating plants and $^1/_{10}$ to $^1/_{20}$ in evaporating plants with thermo-compressors. These considerations apply not only to electrically-driven heat pumps, but also to any form of drive such as steam turbines (Figs. 16—19) or internal combustion engines: *in every case, the heat pump ensures a saving of high-grade energy (electricity, fuel) equivalent to several times the amount expended.*

Conservation of energy resources, no matter whether they be in form of fuel or electricity, is no transient necessity, but one of the most urgent problems of the future.

Coal tends to become more and more a raw product rather than a fuel. Refining plants between the pit and consumer[2] extract valuable base products for the ever-increasing number of synthetic substances manufactured by the chemical industry, whereas the residual coke is available as fuel. There are also good reasons for anticipating[3], that after the war, the prices of fuel will not return immediately to low and lowest levels.

Finally, the limited supply of energy together with the law of supply and demand render it necessary to differentiate the claims to hydro-electric energy of various classes of consumers[4]. Recent restrictions in winter have shown up clearly the limitations of this indigenous supply of energy, particularly with regard to heating. Consequently, the need of making the most of available resources is likely to subsist.

Bearing these considerations in mind, the possibilities of the heat pump will have to be carefully examined in the future whenever conditions are suitable and whenever heat in the form of hot water or hot air has to be produced at temperatures not exceeding 80 to 100 ^{0} C, or the heat contained in water vapour which is given off, as is the case, for instance, in evaporating plants, can be recuperated.

From the point of view of national economy, the heat pump is of interest inasmuch as it not only enables fuel to be replaced by inland sources of heat, but also, as has been pointed out by Professor Bauer[5], because it ensures a more rational consumption of electricity

[1] See "Electricité, miroir de la technique moderne à l'Exposition Nationale Suisse 1939", page 120, published by "Electrodiffusion" Société Suisse pour la diffusion de l'énergie électrique.

Furthermore: "Vom Temperaturwert der Kalorie", Rundschau Deutscher Technik, February 18th, 1943, p. 4.

[2] Thau: "Brennstoff und Wärmewirtschaft", Nos. 2, 26 and 29, 1942.

[3] Dr. Ad. Meyer: "Post-war Steam Power Plant", p. 131 of the present number of this journal, and Prof. Dr. B. Bauer: "Brennstoffeinsparung in der Industrie mit der Wärmepumpe", Elektrizitätsverwertung, No. 1, 1942/43.

[4] Prof. Dr. B. Bauer: "Die Wärmepumpenanlage Walcheplatz in Zürich", Technical Supplement No. 406 of the Neue Zürcher Zeitung, March 10th, 1943.

[5] "Anpassung der Wärmeversorgung der Schweizerischen Industrie an die gegenwärtige und kommende Kohlenwirtschaft", p. 12. This is the publication of a series of short papers presented at the general meeting of the Swiss Association of Energy Consumers at Zurich, on March 24th, 1942.

for producing heat. With the heat pump, as opposed to the electric boiler, the greater part of the heat consumed is available on site in the form of heat contained in the surroundings, with the result that the load on the electric plant (turbines, generators, transformers, and transmission lines) is reduced accordingly, and a greater margin of high-grade electrical energy thus becomes available. It is also for this reason that the heat pump belongs to the electrification programme of industrial heating.

II. WHAT CONDITIONS HAVE TO BE FULFILLED BY A HEAT PUMP FOR HEATING?

(a) *Either heat from the surroundings or waste heat must be available in sufficient quantities.*

(b) *The temperature difference between the available source of heat and that required for heating must not be too great,* if possible, not more than 50—80 ° C.

The conditions (a) and (b) are those necessary to obtain *high coefficients of performance** for the heat pump, as it only then that it is worth while utilizing heat from the surroundings or going to waste.

(c) *The heating capacity must be big enough.* According to the present stage of technical development, it should amount to at least 150 000 — 300,000 kcal/h**. This applies particularly to auxiliary vapour heat pumps with turbo-compressors.

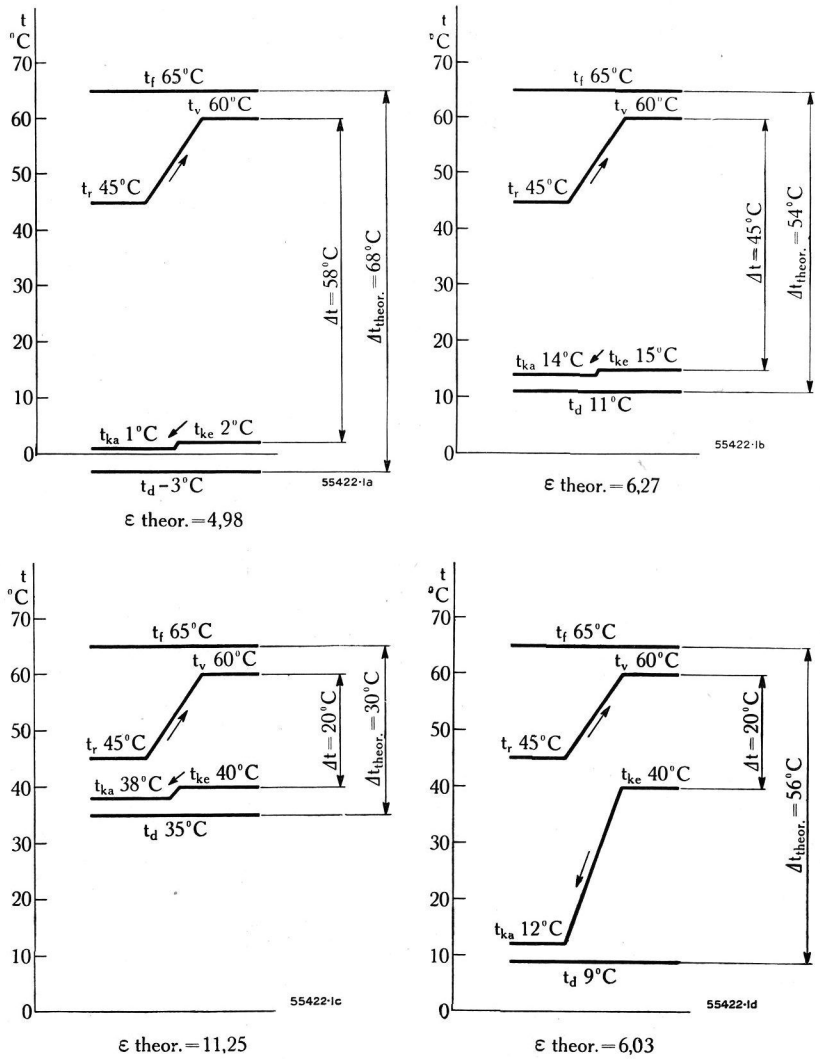
(d) *The number of operating hours per year and the degree of utilization* of the plant *should be as high as possible.*

(e) If an electric drive comes into consideration — which is mostly the case — the *necessary electrical energy must be available at a reasonable price.*

The various conditions just outlined will be now examined in somewhat greater detail.

* See A. Meldahl: "The Heat Pump as Refrigerating and Heating Machine", The Brown Boveri Review, 1943, p. 82.

** 1 BTU = 0·252 kcal.



Figs. 1-4. — Temperature diagrams for heating water from 45 ° to 60 ° C for four different conditions of the source of heat.

Figs. 1 and 2.

The source of heat consists of river water at 2 ° C during winter (Fig. 1) and 15 ° C during summer (Fig. 2). As an ample amount of river water is available, it follows that its temperature drop across the evaporator ($t_{ke} - t_{ka}$) is small, which ensures a comparatively high performance coefficient ε.

Figs. 3 and 4.

Fig. 3 shows the conditions when waste water at 40 ° C is available in ample quantity, whereas in Fig. 4 it is limited. Despite the high temperature of the source of heat, high performance coefficients ε can be obtained only if waste water is available in sufficient quantity (Fig. 3), otherwise its temperature drop across the evaporator can become so great that the performance coefficient is no higher than if the evaporator is heated with 15 ° C river water (Figs. 4 and 2).

Among the conditions necessary to obtain a high performance coefficient for the heat pump it is necessary to ensure an ample supply for the source of heat.

t_v. Temperature of hot water leaving condenser (60 ° C for all cases).

t_r. Temperature of hot water returning to condenser (45 ° C for all cases).

t_{ke} and t_{ka}. Temperatures of the source of heat when entering and leaving evaporator.

Δt. Temperature difference between hot water leaving condenser and source of heat entering evaporator.

t_f. Liquefaction temperature of auxiliary vapour in condenser.

t_d. Evaporation boiling temperature of auxiliary vapour in evaporator.

$\Delta t_{theor.}$ Temperature difference ($t_f - t_d$).

ε theor. Performance coefficient, defined as follows

$$\frac{(273 + t_f)}{(273 + t_f) - (273 + t_d)} = \frac{(273 + t_f)}{\Delta t_{theor.}}$$

Regarding IIa and IIb: *Sources of heat and coefficient of performance.*

Practically unlimited sources of heat are constituted by water from rivers and lakes; ground water, air from rooms or out of the interior of the earth, such as from deep mines can also come into consideration, as well as hot air saturated with steam given off during manufacturing processes, etc. An interesting source of heat is also afforded by warm cooling water from machines, apparatus, etc., the temperature of which is not high enough to permit of direct utilization of its heat, which would otherwise represent a more simple solution. However, it is essential if waste heat is to come into consideration as a source of heat that it should be available in ample quantities. Otherwise the temperature drop in the evaporator becomes so great as to impair the coefficient of performance. The influence of the magnitude of the source of heat is shown clearly by Figs. 1—4, a comparison between Figs. 3 and 4 being particularly instructive. If the same results are obtained from waste heat and river water, preference should be generally given to the former as a source of heat because as a rule its temperature is not liable to so many fluctuations throughout the year, and the intake is more simple than if water has to be drawn from lakes or rivers.

Whenever water from the surroundings is taken as a source of heat, the heat-pump plants have to be located in the vicinity of streams, rivers or lakes in order to keep down the costs of pumping. As a norm for the quantity of water required, it can be stated that to *one m^3 of water per second flowing through the evaporator corresponds an hourly heating output of about 4·5 Mio*[1] *kcal,* it being assumed that the lowest temperature of the source of heat during the winter amounts to 2 ° C and that the hot water leaves at a temperature of 60 to 70 ° C.

Small and medium-size plants can eventually make use of the drinking-water supply as a source of heat, since during the winter and intermediate seasons when heating is required, the capacity of the waterworks is generally in excess of the normal demands. In this manner the load factor of the waterworks is improved and, at the same time, the heat pump no longer requires a special pumping station with its attendant pipelines. Furthermore, during the winter the temperature of this water is generally higher than that of fresh water from lakes or rivers, which has a favourable effect on the coefficient of performance.

At first sight, atmospheric air, which is available everywhere, would appear to afford a convenient source of heat. Its utilization for this purpose, however, although possible in principle, is subject to a number of drawbacks. The temperature of the source of heat should remain constant within fairly narrow limits, otherwise the heating output and leaving temperature are both subject to prohibitive fluctuations. Furthermore, the precipitation of water contained in the air can give rise to operating difficulties as soon as its temperature drops below freezing point, due to deposits of frost or even ice on the surfaces of the evaporator. Unless special precautions are taken, the use of atmospheric air as a source of heat is therefore not to be recommended.

The effective coefficient of performance ε_{kl} referred to the power intput at the motor terminals can be written as follows:

$$\varepsilon_{kl} = \frac{Q}{AL} = \frac{T_v}{T_v - T_{ke}} \cdot \eta = \frac{T_v}{\Delta t} \cdot \eta \quad \text{where}$$

Q designates the heating output in kcal/h.

AL the thermal equivalent in kcal/h of the electrical energy expended by the driving motor.

T_v leaving temperature to heating system in ° K.

T_{ke} temperature of the source of heat in ° K (e. g., river water).

Δt the difference between the two aforementioned temperatures.

η the overall efficiency of the installation[2] due to the various losses, including the irreversibility of the heat-pump cycle.

For auxiliary-vapour heat pumps with turbo-compressors, the overall efficiency varies approximately as follows:

Heating output of the unit kcal/h	Overall efficiency η
200,000—1,000,000	0·45—0·55
1,000,000—3,000,000	0·55—0·60
over 3,000,000	0·60—0·65.

From the foregoing it will be perceived that — for a given temperature of the source of heat — the performance coefficient becomes higher as the difference between the leaving temperature to the heating system and the temperature of the source of heat becomes smaller, i. e., the temperature difference which has to be surmounted by the heat pump becomes smaller (c. f. Figs. 1 and 2). The following performance coefficients are influenced by this temperature difference, the lower figures corresponding to small installations and low temperatures of the source of heat (river or lake water in winter), whereas the higher figures are valid for larger installations and favourable temperatures for the source of heat (river or lake water during the intermediate seasons and summer, ground and tepid waste water).

[1] Mio is the abbreviation for million.

[2] For further particulars regarding the signification of the overall efficiency, see A. Meldahl: "The Heat Pump as a Refrigerating and Heating Machine", The Brown Boveri Review 1943, p. 83.

Application.

	Performance coefficient referred to motor terminals
1. *Room heating* with standard radiators for 90/70 ° C, leaving temperature from heat pump 65 ° C	2·5—4
2. *Room heating* with radiators for 45 °C leaving temperature from heat pump, heating by radiation or from ceilings	3·5—6
3. *Heating hot water* to 70 ° C for industrial purposes:	
(a) with water from surroundings . .	2·5—4·0
(b) with warm waste water	3·0—7·5
4. *Swimming pools.* Leaving temperatures from heat pump 40 ° C	
during winter	4·0—6·0
during summer.	6·0—8·0
5. *Air heating.* Leaving temperature from heat pump 30 ° C	
(a) with closed circuit (Fig. 26) . .	2·5—6·0
(b) with fresh air and ground water .	8—12
6. *Evaporating plants* with thermo-compressor	
for solutions boiling with difficulty .	5—10
for solutions boiling readily . . .	11—25

Power input.

The foregoing figures enable the approximate electrical power input and motor rating to be easily determined, as can be seen from the following example:—

For a case according to 2, the heating output amounts to 3·0 Mio kcal/h and the coefficient of performance to 4·5. The power input at the motor terminals is therefore:

$$L = \frac{3{,}000{,}000}{4{\cdot}5 \times 860} = \underline{775 \text{ kW}}$$

i.e., 4·5 times smaller than with direct electric heating.

Fig. 5 shows the performance coefficients ε_{kl} which can be obtained with large heating units producing hot water with the temperature of the source of heat as a parameter.

Regarding II c: *Influence of the heating output per heat-pump unit.*

Increasing the output per unit ensures lower specific installation costs of the heating plant (Fig. 6), and at the same time, a more favourable overall efficiency and performance coefficient for given operating conditions. In other words, the economy is thus increased.

The lower limit of the output per heat-pump unit in the case of an auxiliary vapour machine with turbo-compressor lies around 300,000 to 150,000 kcal/h, according to the temperature of the source of heat.

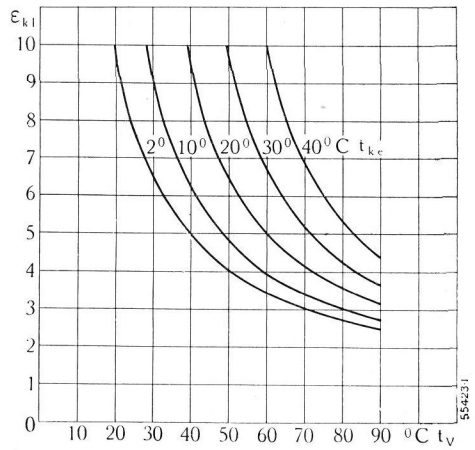


Fig. 5. — Performance coefficients of heating machines (auxiliary vapour heat pump with turbo-compressor) of large capacity for producing hot water, the source of heat being water at different temperatures taken from the surroundings.

t_v. Hot-water leaving temperature.

t_{ke}. Temperature of source of heat.

ε_{kl}. Performance coefficient of heat pump referred to input at motor terminals (for definition see text; overall efficiency $\eta = 60\,\%$).

This diagram shows the performance coefficient ε_{kl} for a given leaving temperature t_v of the heating system and for different temperatures t_{ke} of the source of heat. For other overall efficiencies a corresponding correction is necessary.

A performance coefficient of 6, for instance, means that with an expenditure of one kWh, it is possible to produce an amount of heat equal to six times the thermal equivalent of one kWh, viz., $6 \times 860 = 5160$ kcal.

Therefore, the heat pump should be considered whenever electricity is used for heating, because in this way not only coal but also electricity is saved.

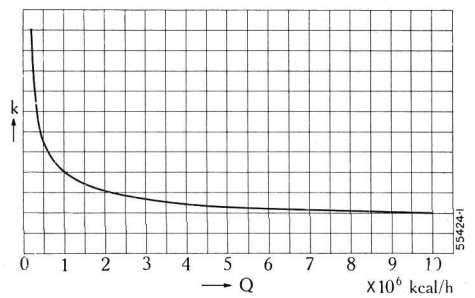


Fig. 6. — Specific installation costs of heating machines.
(Auxiliary vapour heat pump with turbo-compressor.)

k. Specific installation costs. Q. Heating capacity in Mio kcal/h.

The installation costs increase rapidly for small heating capacities. Auxiliary vapour heat pumps with turbo-compressors come therefore primarily into consideration for medium and large plants, having heating capacities in excess of approximately 150,000—300,000 kcal/h.

According to Bauer and Peter[1] the installed heating output for room heating amounts to 17 to 22 kcal/h per m^3 of enclosed volume for climatic conditions in Northern and Eastern Switzerland, regardless of the influence of the wind and radiation, the conventional outdoor temperature being — 20° C and room temperature 18° C. The smallest built-in volume of buildings that can be heated by a heat pump amounts

[1] B. Bauer and W. Peter: "Wasser und Energiewirtschaft", No. 7/8, 1935, p. 109.

therefore to 7000 to 17,000 m^3, according to the atmospheric and climatic conditions coming into consideration. For instance, if the minimum outdoor temperature is higher or if lower room temperatures have to be maintained — e. g., in workshops, garages, shops, etc. — or if the heat pump delivers only a fraction of the total heating output, the minimum enclosed volume is increased accordingly.

Regarding II d: *Load factor and number of full-load operating hours.*

As with any other machine, these factors have in the present case an appreciable influence on the cost of the heat produced. They can be defined as follows:—

Load factor of the heating machine:

$$a = \frac{\text{amount of heat effectively produced per year (kcal)}}{\text{maximum amount of heat that can be produced per year (kcal)}}$$

Number of full-load operating hours:

$$h_{fl} = \frac{\text{amount of heat effectively produced per year (kcal)}}{\text{hourly full load heating output (kcal/h)}}$$

The cost of producing heat becomes smaller as the number of hours per year during which the machine is running fully loaded is increased.

Industrial processes requiring large amounts of heat and operating continuously naturally give the most favourable results. The conditions for room heating are examined in Chapter IV and Table I.

Regarding II e: *Cost of electrical energy.*

The supply of current for heating during peak periods generally offers little interest to the supply undertakings on account of the low tariffs based on equivalence with the cost of fuel. On the other hand, with the heat pump which converts heat from the surroundings against an expenditure of energy which is the equivalent of only a fraction of the amount of heat produced (see performance coefficient), it is possible to pay higher prices for electric energy. These are not proportional, however, to the performance coefficient, because the first costs of a heat-pump installation are considerably higher than those of a corresponding installation heated by fuel. The fixed charges for amortization and interest are therefore affected accordingly, and have an appreciable influence on the cost of the heat produced. The price which can be paid for electrical energy is therefore lower than that obtained only from comparison with the cost of fuel. As the latter increases, the conditions become rapidly more favourable for the heat pump. With coal costing 60 to 80 Fr. and more per ton delivered to the bunkers and given favourable operating conditions, it becomes possible with new plants of large capacity to pay standard rates for electrical energy supplied to motors. Evaporating plants can be written off within a few years, even if the number of operating hours is relatively small.

The conditions for heating machines become still more favourable if they can be adapted to the load

TABLE I, corresponding to the example of a room-heating plant in Section IV.

Numerical example of combined room heating with a heat pump and fuel, for a rated heating capacity of 6 Mio kcal/h.

(a) Heat pump.

Percentage of the rated heating capacity furnished by the heat pump	Rated heating capacity of heat pump	Daily heating output of the heat pump with −20° C outdoor temperature	Percentage of annual heat consumption furnished by heat pump	Number of full-load operating hours of heat pump	Warming up	Normal operation
					The heat pump alone is sufficient for outdoor temperatures not lower than	
	Mio kcal/h	Mio kcal		h approx.	°C approx.	°C approx.
50	3·0	72·0	99·0	2060	— 1·7	—10·5
40	2·4	57·6	97·5	2520	+ 1·2	— 5·3
30	1·8	43·2	92·0	3200	+ 4·0	— 0·6

(b) Heating with fuel alone.

Total amount of heating necessary with outdoor temperature $t_a = -20°$ C	Total amount of heat required when $t_a = -20°$ C	Load factor referred to an operating period of 225 days	Number of full load operating hours when heating with fuel alone
Warming up 6·0 Mio kcal/h Normal operation 4·0 Mio kcal/h	102 Mio kcal	28 %	about 1030

conditions on the electrical network, so as to obtain tariff concessions. For instance, contracts can be made with the power-supply undertaking without obligation to deliver energy at all times, with the implication that after previous notification, the heat pump has to be shut down during temporary periods of shortage of electrical energy, it being then necessary to resort to fuel for maintaining the heating. Experience shows that with hydro-electric power plants, a shortage of water necessitating a curtailment of the supply of electrical energy occurs so exceedingly seldom in normal years, as to have practically no influence on reducing the saving in fuel and load factor of the heat pump.

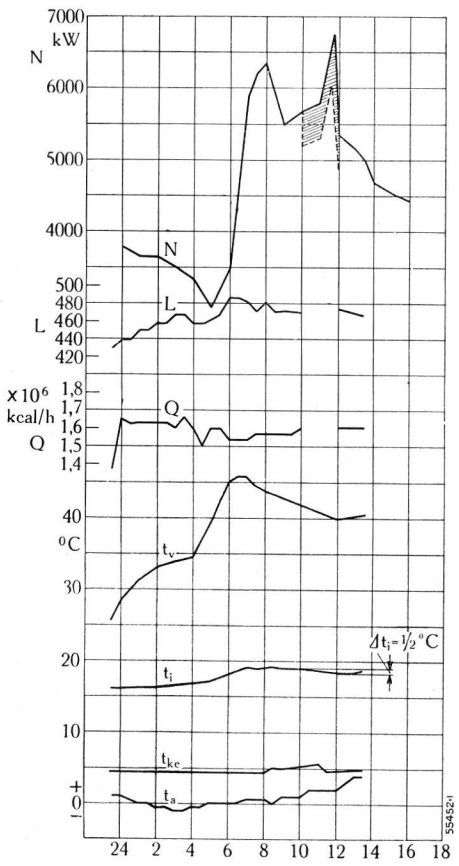


Fig. 7. — Daily load diagram of a heat pump for room heating with on-and-off regulation.

Abscissae: Hours of the day.

Ordinates: *N*. Load on electric network in kW.
L. Power taken by heat pump from electric network in kW.
Q. Heating capacity of heat pump in kcal/h.
t. Temperatures.
t_v. Hot water leaving temperature in ° C.
t_i. Room temperature in ° C.
t_a. Outdoor temperature in ° C.
t_{ke}. Temperature of source of heat in ° C.
Δt_i. Drop in room temperature, during stoppage of heat pump (in the present case ½ ° C per hour).

The shaded surface shows how the load on the electric system can be reduced during the cooking peak by stopping the heat pump.

In order to obtain favourable load conditions for the power-supply undertaking, the warming up of rooms (possibly a hot-water accumulator) should be carried out with fully-loaded heating machine (best efficiency) during the night or off-peak periods. The heat pump is shut down when the maximum load on the electric network occurs or the room temperature is sufficiently high.

For room heating it is also possible to avoid daily load peaks (e. g., cooking peaks between 10 a. m. and noon) by shutting down the heat pump during these periods (on-and-off operation). Fig. 7 shows in a very simplified manner the superimposition of a heat pump on the load diagram of a municipal electric system. The heat-storage capacity of the heated rooms, offices and workshops plays a very important part if the heat pump has to run intermittently, since on it depends the temperature drop during the shut-down periods. For instance, experience shows that a heat drop of 2° C can be perfectly well allowed for domestic heating plants.

If, on the other hand, it is essential to maintain the leaving or room temperature constant, — as is the case, for instance, for heating processes or heating of modern buildings which are very pervious to cold, — uninterrupted heating becomes necessary. The addition of a heat accumulator enables load peaks of the electrical network to be bridged over in such cases.

The consumption of energy and operating time in air-conditioning plants can be considerably increased by utilizing the same heat pump for heating in winter and cooling in summer. For these reasons installations of this description are likely to be viewed with favour by power supply undertakings, who will be prepared to sell energy at advantageous tariff rates.

III. INDUSTRIAL APPLICATIONS OF HEATING MACHINES.

Industrial applications are by far the most interesting for heat pumps which can come into consideration whenever large amounts of hot water or air are required. Suitable conditions occur for instance in the textile industry (rayon and staple fibre factories, dye works and finishing plants, spinning and weaving mills), in paper and chemical works, as well as in plants like slaughter-houses, breweries, large laundries, tanneries, etc. The heavy demands for heat in installations of this description together with the large number of operating hours ensure good utilization of the heat pump and favourable tariff conditions.

Although in a great many cases, effective temperatures of 80° C are not exceeded, heating is commonly effected by steam under a pressure which may be several times that of the atmosphere. However, plants of this description can be frequently rebuilt without much expense for hot-water heating, and rendered suitable for operation by means of a heat pump.

Drying processes frequently lend themselves particularly well to the heat pump, especially when use can

be made of the waste heat, such as drying brewery malt in kilns, felts in paper works, simple or twisted yarn in sizing machines, dehydrating foods and chemical products, drying all kinds of materials in drying rooms, vacuum drying cubicles, pulverizing towers, etc.

The multiplicity of drying processes in the various branches of industry renders it impossible to lay down hard and fast rules as to the adaptation of the heat pump. Each case has its own special problems depending on the drying temperature, speed and potential as well as the degree of humidity necessary for carrying out the drying process. Liquids, for instance, are frequently treated in evaporating plants in conjunction with vacuum drying cubicles or pulverizing towers making use of thermo-compressors.

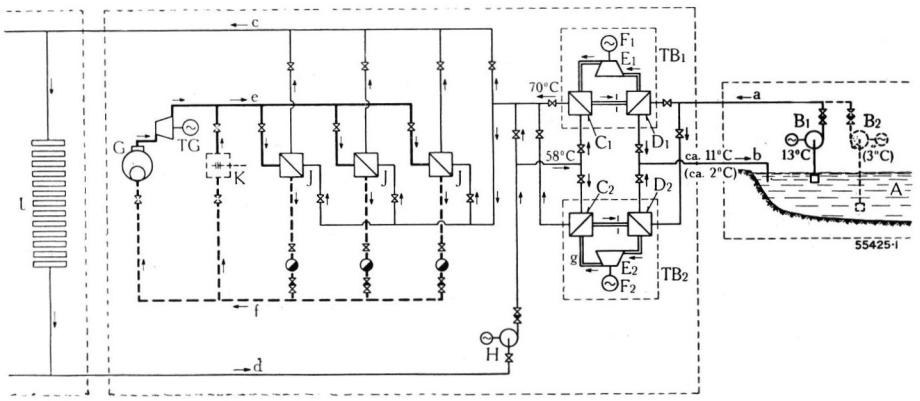


Fig. 8. — Layout diagram of the hot-water plant of the Steckborn Artificial Silk Co., Ltd., Steckborn (Switzerland) with heat pumps as heating machines and supplementary boilers (heated by fuel and electrically).

In order to be independant of coal supplies, heat is obtained in this plant from the Lake of Constance, the temperature of which varies between 3° C in the winter and 13—20° C in the summer, for producing 70° C hot water. Two heat pumps (thermoblocs TB_1 and TB_2) are installed, each producing 1·0—1·7 Mio kcal/h, which correspond to the base heating load. Additional heat for covering peaks is furnished by the boilers G, and at a later date it is intended to install an electric boiler K. The steam produced is utilized in a back-pressure turbine TG, before it gives up its heat in the heat exchangers.

A. Lake of Constance.
B_1. Pump set for summer operation (13—20° C).
B_2. Pump set for winter operation (3° C).
TB_1 and TB_2. Thermoblocs (heat pumps), each comprising:
C. Condenser.
D. Evaporator.
E. Turbo-compressor.
F. Electric motor.
G. Coal-fired boiler.
K. Electric boiler (projected).
H. Hot-water circulating pump.
J. Steam-water heat exchanger.
L. Hot-water consumers.
TG. Turbo-alternator set with back-pressure turbine.
a. Lake-water intake pipeline.
b. Lake-water return pipeline.
c. Hot-water leaving main.
d. Hot-water return main.
e. Heating steam.
f. Pipeline for condensed heating steam.
g. Working-medium circuit.

(a) *Example of an industrial heat-pump plant for producing hot water (Steckborn Artificial Silk Co., Ltd., Steckborn, Switzerland).*

For manufacturing rayon, a considerable amount of hot and warm water at low and medium temperature is required for heating the spinning baths, spinning machines, drying ducts and basins as well as for the spun material itself.

The conditions at Steckborn are particularly well suited for a heat pump since a practically unlimited heat source is provided by the Lake of Constance; also the demands for heat are considerable and practically constant. Moreover, the plant is in continuous operation throughout the year, thus ensuring a high load factor.

Steam was used formerly for heating and the plant was re-adapted for hot-water heating, so as to render its operation as independent as possible of fuel. At the same time, considerable advantages were secured due to the cleanliness of the heat pump, as well as to the elimination of steam and condensed water, with the attendant troubles inherent in leaky steam traps and joints, etc.

As can be seen in Fig. 8, heat is now produced by:

Two thermoblocs, which maintain the base load for hot-water heating.

Boilers for processing steam and for covering peaks for hot water heating. At the present time, coal-fired boilers are installed, but it is proposed to add in the future an electric boiler. Heating is effected by means of steam-water heat exchangers.

One of the two heat pumps (Fig. 9) is sufficient at present for normal requirements, whereas the other serves as standby and for covering requirements after contemplated extensions to the plant have been carried out. The steam produced by the boilers is first of all expanded in a back-pressure turbine, to a pressure suitable for heating apparatus requiring heat at relatively high temperatures. This low-pressure steam is also used for heating the hot-water system connected to the heat pumps during peak periods and whenever there is a shortage of electrical energy due to low water.

The intake of water from the Lake of Constance is at a certain distance from the shore, and is arranged so that water is always taken in at as high a temperature as possible, — viz., from the surface during the summer and from a deeper level during the winter, — in order to secure the maximum performance coefficient obtainable. The pumping station comprises two vertical-shaft, bore-hole pumps located in a special

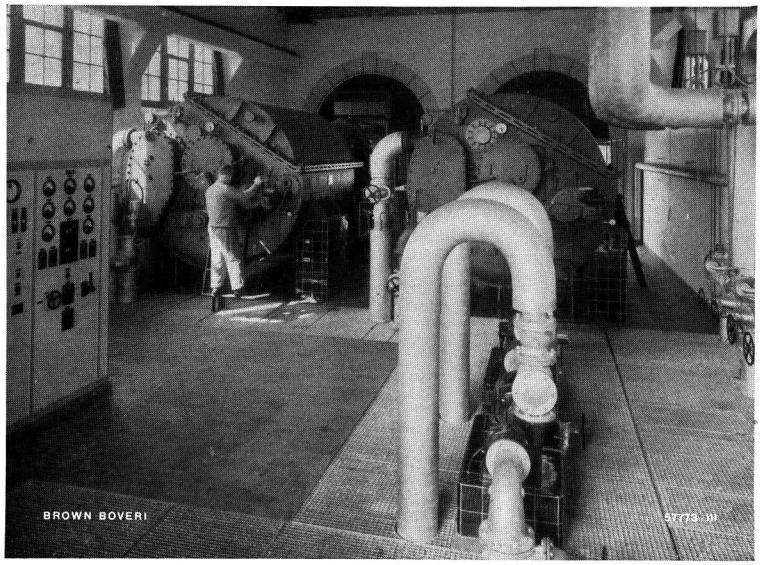


Fig. 9. — Heat-pump plant of the Steckborn Artificial Silk Co., Ltd., Steckborn (Switzerland),

comprising two thermoblocs (heating machines), each for 1·0—1·7 Mio kcal/h. With lake water at 13 °C each unit has a normal heating capacity of 1·51 Mio kcal/h when heating hot water from 58° to 70° C. About two thirds of this amount of heat is extracted from the water of the Lake of Constance, whereas the remaining third is furnished by the electrical energy consumed.

With one unit alone in operation, roughly 2100 tons of coal per year are saved.

building. They have been dimensioned for delivering different quantities of water, the larger pump being intended for winter operation when low water temperatures occur and the temperature drop in the evaporator has to be kept as small as possible, whereas the smaller one is for summer operation or when there are reduced heating loads.

The adoption of thermoblocs, incorporating smooth-running turbo-compressors greatly simplified the design of the foundations on unstable alluvial soil. Each thermobloc rests on four small concrete pillars and the load is evenly distributed over the foundation piles by means of a sole-plate. In addition to two thermoblocs and three steam-water heat exchangers, the building also contains two motor-driven hot water pumps, the governing devices and the switchgear.

Each thermobloc consists of a turbo-compressor, driven by an 8000 V induction motor over gearing, an evaporator with spray pump for the working medium, and a condenser (Fig. 10). Methylene chloride ($CH_2 Cl_2$) has been adopted as working medium. Special attention must be drawn to the three-speed gear which enables the compressor speed, and consequently the heating output to be regulated according to the seasonal fluctuations of the temperature of the lake water. A switchboard and high-tension cubicles contain the switchgear and instruments necessary for operating and controlling the plant.

The automatic governing devices maintain the leaving temperature of the hot water constant and ensure at the same time that as much heat as possible is always supplied by the thermoblocs, both when these machines are running alone and in conjunction with the steam-water heat exchangers.

(b) *Example of a heat pump for industrial drying purposes (Landquart Paper Mills, Landquart, Switzerland).*

This installation is of interest inasmuch as the heat pump operates with *air* and not with an auxiliary working medium. This design is specially suited for cases where air must be heated or whenever an insufficient amount of water is available.

Air heat pumps require not only a compressor but also an expansion machine[1]. These two machines are combined in an ingenious manner in the Brown Boveri-Lèbre cellular rotor.

[1] A. Meldahl: "The Heat Pump as a Refrigerating and Heating Machine", The Brown Boveri Review 1943, p.75–85.

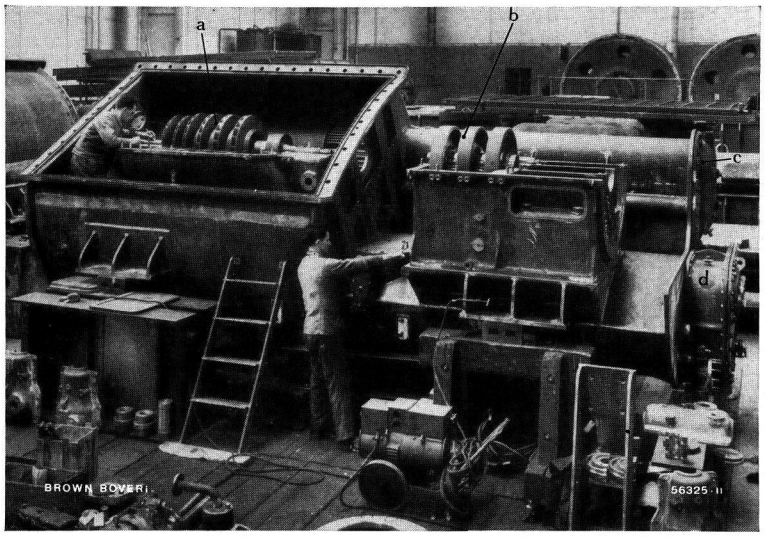


Fig. 10. — View taken during erection in the shops of a thermobloc for the Steckborn Artificial Silk Co., Ltd., Steckborn, of 1·0—1·7 Mio kcal/h heating capacity for producing hot water.

a. Turbo-compressor.
b. Partly-erected gear between turbo-compressor and driving motor.
c. Condenser.
d. Evaporator.

Attention is drawn to the three-speed gear b, which enables the compressor speed, and consequently the heating output, to be regulated according to the seasonal fluctuations of the lake-water temperature.

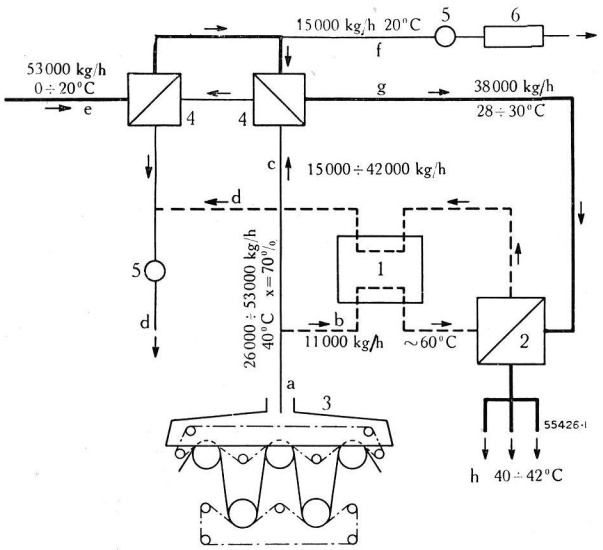


Fig. 11. — Layout diagram of the heat pump plant used for drying in the Landquart Paper Mills, Landquart (Switzerland).

1. Cellular rotor } of the
2. Heat exchanger (air heater) } Brown Boveri-Lèbre air heat pump.
3. Vapour collector of the paper machine.
4. Two-stage recuperation plant for heating fresh air from 0—20° to 30° C.
5. Fans.
6. Heater for air used for defogging.

a. Air exhausted from paper machine (40°C, relative humidity x = 70%).
b. Branch of exhaust air used in heat pump as source of heat.
c. Branch of exhaust air used in recuperator 4 for heating fresh air.
d. Cooled air to atmosphere.
e. Fresh air.
f. Branch of warm fresh air for heating machine room.
g. Fresh air heated to 30°C going to heat exchanger 2 of air heat pump.
h. Warm air at 40—42°C for drying felts.

The first air heat pump with cellular rotor in Switzerland was installed for conditioning air in one of the halls of the Congress Building in Zurich by the heat-pump research committee of the Swiss Federal Institute of Technology together with Brown Boveri.[1]

The Landquart Paper Mills is the first instance of an air heat-pump with cellular rotor being installed for an industrial heating process. Various far-reaching improvements to the air-flow conditions through the cellular rotor were incorporated with the result that a coefficient of performance of $\varepsilon_{kl} = 2{\cdot}78$ was obtained with the operating conditions coming into consideration. Subsequent research has shown that the air heat pump with cellular rotor can be still further improved and cheapened. It can consequently come into consideration for a number of industrial applications for which its very simple operation — air as operating medium, no evaporator but only a heat exchanger serving as air preheater on the hot side in the place of a condenser — renders it eminently suitable.

The plant in Landquart is designed for a heating output of 115,000 kcal/h. Moist, warm air coming from a paper machine serves as source of heat, and is also utilized in waste-heat recuperators for heating the fresh air for drying the felts before it traverses the heat exchanger of the heat pump. The heat pump enables approximately 180 tons of coal to be saved per year, without taking into account the preheating in the waste-heat recuperators. A welcome ancillary feature is the considerable lengthening of the life of the felt strips used in the paper machine due to the increased temperature of the drying air.

Fig. 11 shows diagramatically the layout of the plant with heat pump and recuperators for fresh air and Fig. 12 the heat pump itself.

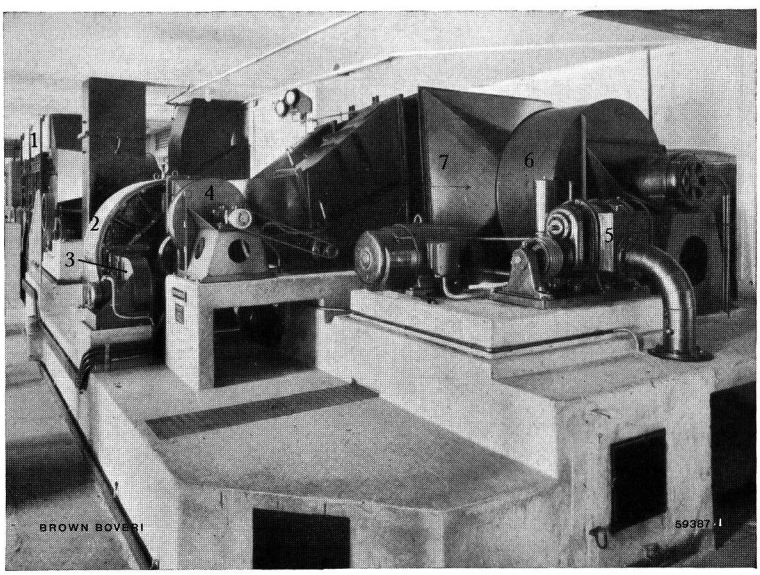


Fig. 12. — 115,000 kcal/h air-heat pump installation of the Landquart Paper Mills, Landquart (Switzerland).

In this interesting installation an air heat pump is used for heating, the compression and expansion being carried out by means of a cellular rotor according to the Brown Boveri-Lèbre system. The source of heat consists of the moist, warm air leaving the paper machine. The heat pump receives air which has been already heated to 30° C in recuperators, and raises its temperature to 40° C, before it is used for drying felts in the paper machine. This relatively small heat pump alone enables approximately 180 tons of coal to be saved per year.

1. Paper machine
2. Housing of cellular rotor } of the Brown Boveri-Lèbre air heat pump.
3. Low-pressure scavenging fan } of the Brown Boveri-Lèbre air heat pump.
4. High-pressure scavenging fan } of the Brown Boveri-Lèbre air heat pump.
5. Additional compressor.
6. Fan for heated fresh air.
7. Heat exchanger of the air heat pump (only the connecting funnel is visible).

[1] B. Bauer and Bolomey: "The Generation of Heat and Cold Air in Air Conditioning Plants by means of an electrically-driven Heat Pump." Elektrizitätsverwertung, No. 9/10, 1939/40, and also Schweizerische Bauzeitung, Vol. 114, July 1st, 1939, p. 12.

IV. THE HEAT PUMP FOR ROOM HEATING.

For economic reasons (see paragraph IIc, page 149) it is necessary to concentrate heat production in units of average or large capacity.[1] For this reason the heat pump is eminently suitable for heating factories, building blocks, building estates, complete towns or districts, swimming pools, etc., both alone or in conjunction with district heating schemes and heating power stations.

For room heating, the amount of heat delivered is the main product, which must be supplied at reasonable prices. In order that a plant of this description can pay, it is therefore necessary to devote particular care to the dimensioning of the heat pump and heating system, both as regards heating capacity and value of the leaving temperature of the hot water. *As a general rule, heat pumps for room heating are dimensioned for a given base load,* whereas the peaks are covered with the aid of less expensive heat producers, such as boilers heated by fuel or by electricity. If the size of the plant and heating programme have been determined, — i.e., the heating capacity per hour computed from the frequency curves of the average daily temperature[2], the leaving and return temperatures of the heating water, the temperature which has to be maintained in the heated rooms and the average outdoor temperature when heating is stopped, — and the average frequency of the mean outdoor temperatures is known, it is possible to determine how many days a given heating output is needed (surface A, Fig. 13). In this manner, the average number of days per year during which heating is required can be determined.

The base load attributed to the heat pump and the leaving temperature for which it should be designed depend on whether more importance is attached to reducing the fuel consumption, — as is the case at the present time, — or whether in normal times as economical operation as possible is desired. In the latter case, the heating capacity of the heat pump is reduced, thus increasing its load factor and number of full-load operating hours.

[1] Cf. page 149 (Smallest output of heat pumps for room heating).

[2] M. Hottinger "Klima und Gradtage", Springer, Berlin, 1938.

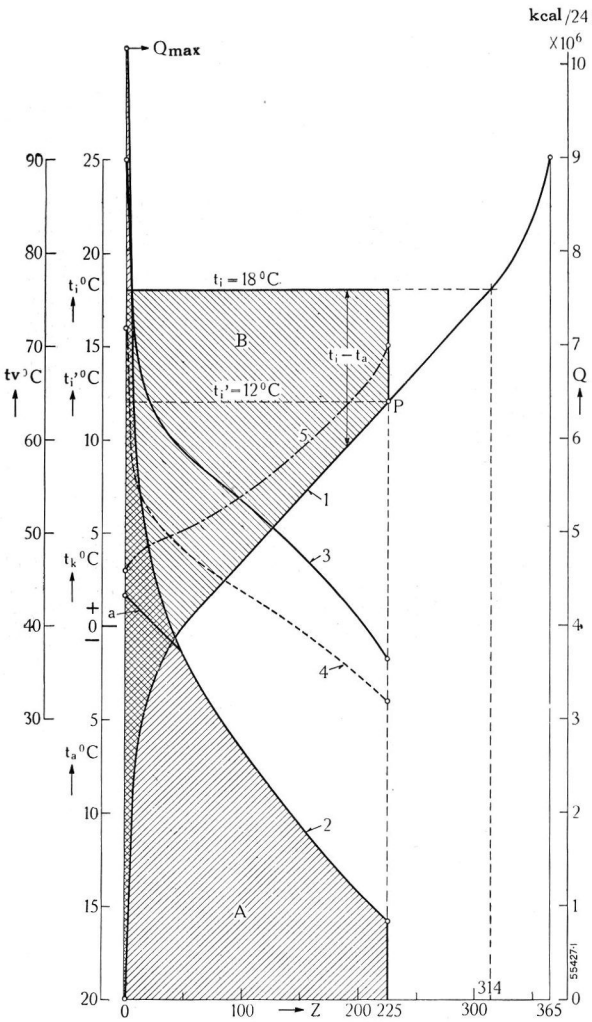


Fig. 13. — **Example showing the determination of the amount of heat required for room heating and the choice of the base load for the heat pump.**

(For climatic conditions at Zurich.)

Abscissae: Z. Number of days per year.

Ordinates: t_a. Mean daily outdoor temperature.
t_k. Temperature of the source of heat (water from the River Limmat).
t_i. Room temperature (18 ° C) to be maintained by heating.
t_i'. Heating limit (12 ° C), i. e., mean outdoor temperature above which heating is stopped.
t_v. Heating water leaving temperature.
Q. Daily heat consumption in kcal deduced from frequency (curve 1) of mean outdoor temperatures.
Q_{max}. Maximum demand for heat in kcal per 24 hours when $t_a = -20$ ° C.

Curve 1. Frequency curve of the mean daily outdoor temperatures in the town of Zurich, according to records of the Swiss Central Meterological Institute between 1869 and 1929. Point P, for instance, implies that in Zurich during the last 60 years, there were on the average 225 days per year with a mean daily outdoor temperature not exceeding 12 ° C.

Curve 2. Frequency curve of the daily demand for heat Q, as deduced from curve 1: $Q = f\,(t_i - t_a)$.

Curve 3. Frequency curve of the leaving temperature t_v of heating water when heating up.

Curve 4. Ditto, for normal operation.

Curve 5. Frequency curve of the temperature t_k of the source of heat.

Surface A. Mean heat consumption in kcal during the entire heating period of 225 days.

Surface B. Mean degree-day surface during the heating period. Each degree Centigrade difference $(t_i - t_a)$ per day, gives one degree-day. If on a given day the difference $t_i - t_a = 6$ ° C, this day reckons as six degree-days. (Cf. Hottinger, "Klima und Gradtage", publishers J. Springer, Berlin, 1938.)

If the heat pump is dimensioned for a base load equal to 30 % of the installed heating capacity, the amount of heat it produces is limited approximately by line a in the surface A. It can cover approximately 92 % of the total yearly demand for heat, only the small amount corresponding to the peaks above line a having to be furnished by additional means. Fig. 14 shows the conditions for other values of the base load.

Line *a* in Fig. 13 bounds the amount of heat furnished by a heat pump designed for 30 % of the installed heating capacity. Table I (page 16) shows in figures the heating conditions for the same installation, operating data with fuel alone having been added for sake of comparison.

It is interesting to note that for the example considered, the heat pump, although dimensioned for only 30 % of the installed heating capacity, can furnish roughly 92 % of the total yearly demand for heat, whereas the remaining 8 % has to be produced by fuel. Fig. 14 shows to what extent the total heat consumption can be covered by the heat pump when it is dimensioned for varying percentages of the installed heating capacity.

The value of the leaving temperature is of decisive importance for the economy of the heat pump. The amount of energy consumed and the rating of the driving motor become smaller as the performance coefficient increases, which, in turn, implies as small a temperature difference as possible between the heating system and source of heat (cf. Fig. 5). Higher leaving temperatures can come into consideration if sufficient warm waste water is available as a source of heat, — e. g., cooling water from compressors or internal-combustion engines, apparatus for finishing processes or in chemical works, or hot waste air, — than if relatively cold water from the surroundings (river or lake water) has to be used for this purpose. In order to obtain as high performance coefficients as possible, it is absolutely essential to choose the leaving temperature as low as can be allowed with respect to the temperature of the water returning from the heating system. Fig. 15 shows in function of the average outdoor temperature, the leaving and return temperatures for standard hot-water heating radiators, designed for 90 ° C leaving and 70 ° C return temperatures when the outdoor temperature is —20 ° C and the installation is being heated up, both for normal heating and when heating up. As can be seen from Fig. 13, curve 4, for the example considered of a heating plant in Zurich, with standard 90/70 ° C radiators and normal heating, leaving temperatures in excess of 60 ° C occur on the average during only five days per year, whereas 50 ° C is exceeded only during 35 days. A maximum leaving temperature of 65 ° C, for example (Fig. 15, curve 6), is sufficient to enable the heat pump to cope with the heating requirements, including losses in the hot-water system and sudden fluctuations of the outdoor temperature, the occasional peaks due to very low outdoor temperatures being met either with additional heating or by increasing the size of the radiators. The heating installation of our works in Baden, which is dimensioned for leaving temperatures of the hot water of only 40 to 45 ° C, constitutes a case in point.

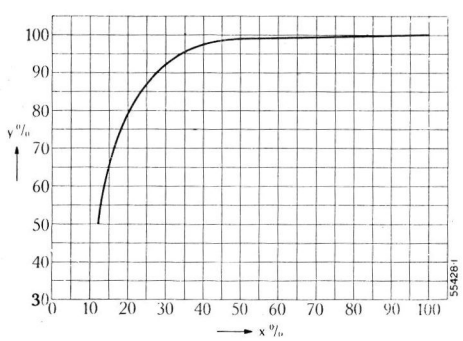


Fig. 14. — Percentage of the total amount of heat consumed per year furnished by a heat pump dimensioned for varying percentages of the total installed heating capacity, the heating conditions being as assumed in Table I.

Abscissae: Heating capacity of heat pump X as a percentage of the maximum heating output of the installation.

Ordinates: Percentage Y of the total yearly consumption of heat furnished by the heat pump.

Heat pumps for room heating have to be dimensioned for the base load. 90 % of the total heat consumption can be covered by a heat pump dimensioned for approximately 30 % of the maximum heating capacity of the plant.

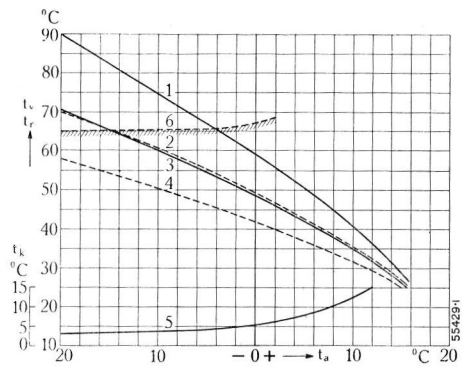


Fig. 15. — Leaving and return temperatures of hot water for standard radiators in function of the outdoor temperature, showing how they can be adapted to operation with a heat pump.

t_a. Outdoor temperature.
t_k. Temperature of source of heat.
t_v. Hot-water leaving temperature.
t_r. Hot-water return temperature.

1 = t_v, 2 = t_r } Temperatures for standard radiators when heating up.

3 = t_v, 4 = t_r } Temperatures for normal operation.

5 = t_k. Temperature of the source of heat in function of the outdoor temperature.

6. Leaving temperatures which can be reached by the heat pump when the temperature of the source of heat t_k varies.

In order to keep the performance coefficient of the heat pump as high as possible and reduce the amount of energy consumed, the leaving temperature of the heat pump should not appreciably exceed 65 ° C. By increasing the surface of the radiators, the heat pump can meet the demands beyond curve 6.

Ceiling heating by radiation is well adapted for heat pumps inasmuch as the hot-water leaving temperature normally amounts to 40—45° C, and does not exceed 55° C.

Still more favourable conditions than with hot water exist when heating is carried out with hot air. In order to eliminate draughts and ensure even distribution of the heat, relatively large volumes of air have to be circulated, with the result that low leaving temperatures are sufficient. These amount to 25—30° C for normal operation, and do not exceed 35° C. These small increases in temperature imply high performance coefficients and a good overall efficiency of the heating plant. This kind of heating is particularly interesting when it can be combined with an air-conditioning installation (cf. page 164).

Before concluding this paragraph, it should be mentioned that heating water for swimming pools is particularly suitable for heat pumps since leaving temperatures not exceeding 40° C are sufficient for normal operation.

Readers are referred to pages 155—156 for explanations concerning the most favourable operating conditions for heat pumps for room heating.

The initial cost is of very great importance in heating plants. The tendency to distribute the heating output over several units, as has been done in particular with reciprocating machines, is frequently due to the desire to utilize available designs rather than to economical considerations. Limitations of this description do not apply to turbo-compressors.

V. THE HEAT PUMP IN CENTRALIZED HEATING PLANTS AND HEATING POWER PLANTS.

Centralized production of heat offers many advantages over heating from a number of small, scattered plants, the chief of which being the following:— More efficient firing; more favourable distribution of load peaks among the various consumers entailing a better utilization of the fuel; suppression of the delivery of fuel to, removal of ashes from, as well as cleaning a number of scattered installations; simplified attendance and supervision of a centralized plant; utilization of fuels which are not suitable for small installations; possibility of purchasing large quantities of fuel at a time at favourable prices; low specific cost of plant due to the installation of only a small number of heating units, etc. The heat losses during transmission and distribution are of secondary importance with respect to all these advantages. Centralized heating is, however, for the time being only an economical proposition for thickly-populated and highly-frequented districts, having big demands for heat.

Burning fuel for producing heating steam at relatively low pressures and temperatures for maintaining rooms at a temperature of 18° C, for instance, represents unwarrantable squandering from the viewpoint of rational utilization of energy. It is far more advantageous against a slightly higher expenditure of fuel to produce high-pressure steam which gives up high-grade energy in a back-pressure heating turbine, the expanded exhaust steam then giving up its heat to hot water in a heating condenser.[1] As the demands for heat do not generally coincide with those for electricity, it is advisable to connect the generators in the heating plant in parallel with a large electrical network, which can absorb the surplus energy and make good any deficiencies.

It now remains to be seen how a heat pump can be fitted into this system comprising fuel, steam, as well as power and heating plants. Is it not a mistake to consume fuel for producing electrical energy which is then employed for driving a heat pump for heating rather than to use the fuel directly for heating? If only countries obtaining electrical energy from fuel are considered, this solution would imply that the demands for power and heat fluctuate together, both with respect to time and location, unless a large-scale distribution of both heat and energy covering considerable tracts of country is resorted to. The latter solution would imply, however, unbearable financial burdens due to the high initial outlay for the heat-transmission pipelines, accumulators, and arrangements for distributing energy, with the result that for the time being it has only an academical interest. On the other hand, a steam-turbine-driven heat pump according to Fig. 19 affords the most rational solution for covering existing demands for heat whenever the simultaneous production of electrical energy does not come into consideration, as is sometimes the case in industrial and urban-heating plants.

The heat-flow diagrams Figs. 16—18 show the percentage of useful heat obtainable from 100 heat units of fuel burnt for other kinds of heating and combined heating and power plants.

The most interesting example is afforded by the utilization of all the available steam for driving the

[1] J. Broggi: "Centralized Heating and Power Plants and the Economics of Heat Recuperation." The Brown Boveri Review, October/November, 1934.

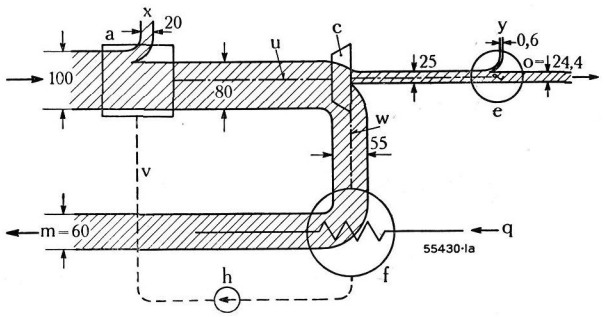


Fig. 16. — Power station with condensing steam turbines in a large city, the energy of which is used in the first place for electric heating* (hot water and steam boilers, radiators, cooking, etc.).

Of 100 kcal introduced as fuel only 24·4 kcal are available in the form of useful energy, which is, however, of very high grade, whereas the greater part is carried away by the cooling water.

* The comparison between the four examples considered is evidently only valid on condition that all energy (high and low grade) produced is used for the same purpose, viz., heating.

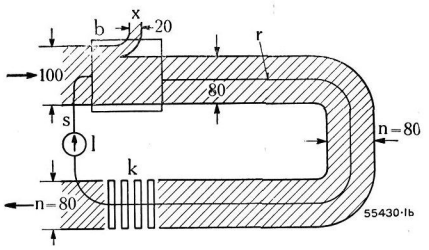


Fig. 17. — Low-pressure heating plant (central heating).

Of 100 kcal introduced as fuel, 80 can be used for heating.

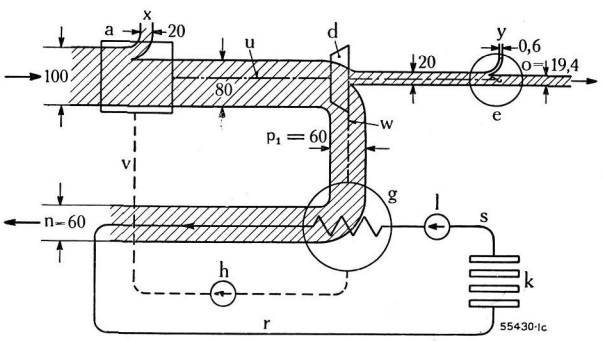


Fig. 18. — Centralized heating power station.

The arrangement differs from Fig. 16 in that *the exhaust* steam from the turbine is not led into an ordinary condenser, but into a heating condenser under a correspondingly higher pressure, where it gives up heat to hot water from the heating plant.

Of 100 kcal introduced as fuel, 19·4 are converted into high-grade energy by the generator, whereas 60 are given up by the heating condenser for heating, so that altogether 79·4 are utilized.

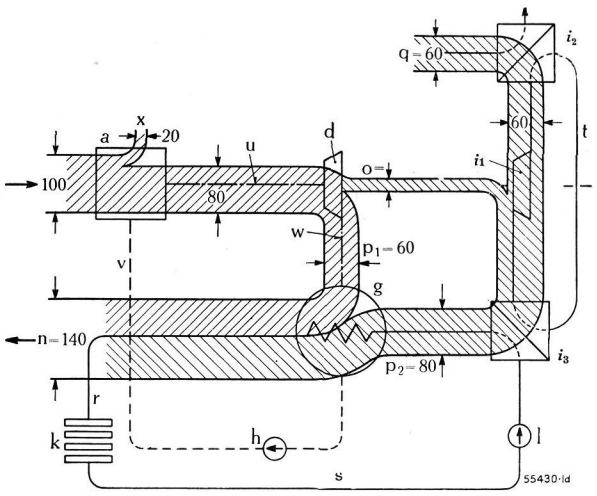


Fig. 19. — Heat pump with steam-turbine drive in a centralized heating station.

The mechanical energy, *equivalent to* 20 kcal, produced by the turbine d is employed directly for driving the heat pump. If the latter has a performance coefficient of 4, it will deliver 80 kcal for heating, whereas a further 60 kcal are supplied by the heating condenser, as in Fig. 18. Hence, an amount of heat for heating of 140 kcal can be obtained from 100 fuel kcal.

Figs. 16—19. — Heat flow diagrams for different kinds of heating, showing the ratio between the amount of heat available for heating and that of the fuel consumed.

The most advantageous results are again obtained with the aid of a heat pump (Fig. 19): from 100 kcal contained in the fuel, 140 kcal can be made available for heating; in other words, 100 kcal for heating require an expenditure of fuel corresponding to only 70 kcal, whereas the remaining heat is taken from the surroundings. These results are interesting inasmuch as they show how large amounts of fuel can be saved whenever considerable quantities of heat are required by the addition of a heat pump.

a. High-pressure boiler.
b. Hot-water heater.
c. Condensing steam turbine.
d. Heating turbine.
e. Generator.
f. Normal condenser, cooled by river water.
g. Heating condenser, cooled by hot water.
h. Feed pump.
i_1. Compressor driven by heating turbine d } of the heat pump (heating machine).
i_2. Evaporator }
i_3. Condenser }
k. Heaters.
l. Hot-water circulating pump.
m. Heat carried away (lost) by cooling water.
n. Useful heat.
o. Thermal equivalent of energy consumed.
p_1. Useful heat from heating condenser.
p_2. Useful heat from heat pump.
q. Cooling water, i. e., source of heat in Fig. 19.
r. Leaving hot-water.
s. Returning hot-water.
t. Working-medium circuit of heat pump.
u. Live steam.
v. Feed water.
w. Exhaust steam.
x. Boiler losses.
y. Generator losses.

heat pump by means of a steam turbine, as shown in Fig. 19 [1].

The production of heat is shared between a back-pressure turbine with heating condenser and the heat pump. For a given heating output, the fuel consumption with this arrangement amounts to only 55 to 60 % of that of a low-pressure heating plant. Speed regulation of the turbine-driven compressor set ensures an efficient regulation of the heating output according to the momentary heating demands. The best possible utilization of the heat content of the fuel is obtained by a Velox steam generator burning pulverized coal,

[1] This arrangement was already mentioned by Lord Kelvin in 1852, and proposed in its present form by Professor B. Bauer, Director of the centralized heating plant of the Swiss Federal Institute of Technology, Zurich.

with which more than 90 % of the available heat in the fuel is utilized, and which can be readily adapted for automatic regulation of the boiler load (Fig. 20).

Similar advantages are obtained by *an electric drive of the heat pump* instead of a steam turbine as in Fig. 19, whenever it is not possible to mount the heating turbine and heat pump together. The overall efficiency with respect to direct drive is only diminished by the electrical transmission losses between the steam turbine and heat pump. Electrically-driven heat pumps are suitable for locating in the centre of gravity of the heat-distributing system, as local heat producers, in thickly populated districts, and can take over with advantage part of the base load. Numerous hygienic and aesthetic advantages are obtained by heating without fuel, such as the elimination of chimneys, with the attendant smoke and ash nuisance. The steam centralized heating and power plant on the other hand can be situated outside the centre of the town, where chimneys and smoke matter less, and its back-pressure turbines will supply the electrical energy necessary for driving the heat pumps, whilst working on the combined heating and power systems.

A typical example of a plant of this description is afforded by *the centralized heating plant of the Swiss Federal Institute of Technology, Zurich,* which comprises a heat-pump station under construction according to the plans of Professor Bauer, which will operate in the manner just described. It is situated in the centre of the town, opposite the main railway station, close to the River Limmat. The latter serves as source of heat, the temperature of which varies between 2 and 16 ° C during the heating period.

Two thermoblocs, the design of which can be seen in Fig. 21, were supplied by Brown Boveri for this plant, each of which is designed for the following operating conditions:—

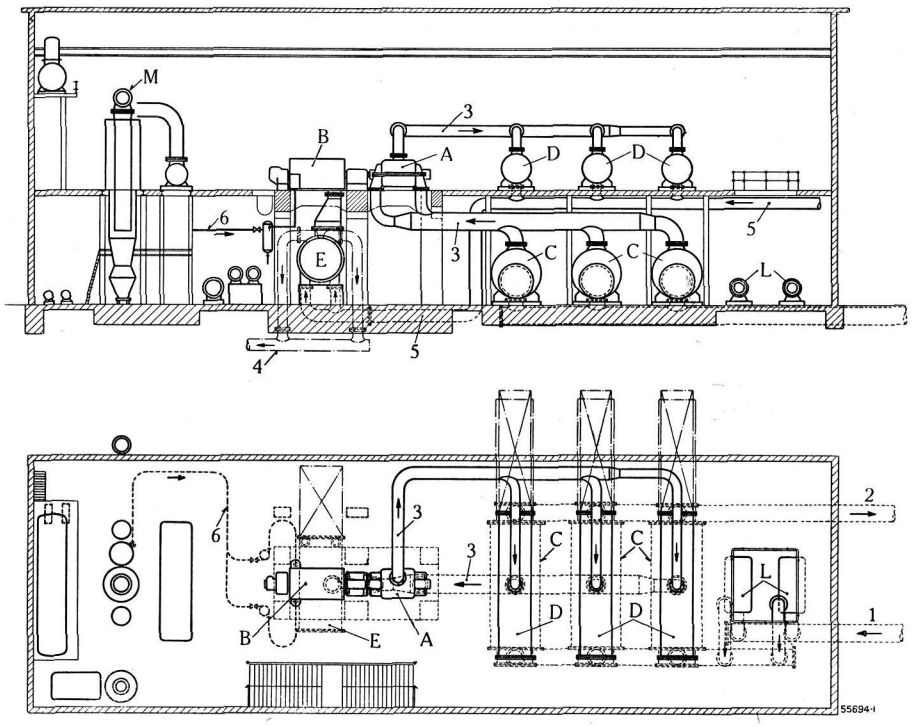


Fig. 20. — Projected heating station for 50 Mio kcal/h with steam-turbine driven heat pump.

This installation operates in the manner shown diagrammatically in Fig. 19, and consists of a Velox steam generator fired with pulverized coal and producing approximately 48 tons of steam per hour. The heating turbine is fed with steam at 100 kg/cm², abs, 450 ° C, and develops at its coupling 9700 kW, which are utilized for the direct drive of the turbo-compressor of the heat pump.

The evaporator and condenser of the heat pump are, on account of their size, subdivided. The amount of water supplying heat taken from the surroundings amounts to about 10,000 m³/h. **Such large units are suitable for centralized plants heating large cities.** An installation of this size would enable approximately 7000 tons of fuel to be saved per year with respect to a low-pressure steam heating plant.

A. Turbo-compressor of heat pump.
B. Heating turbine.
C. Evaporators of heat pump.
D. Condensers of heat pump.
E. Heating condenser.
L. Circulating pumps for water from which heat is taken.
M. Velox steam generator for pulverized fuel.

1. Water inlet (source of heat).
2. Water outlet (source of heat).
3. Working-medium circuit of heat pump.
4. Hot water going to heat consumers.
5. Hot water returning from heat consumers.
6. Live steam pipe to heating turbine.

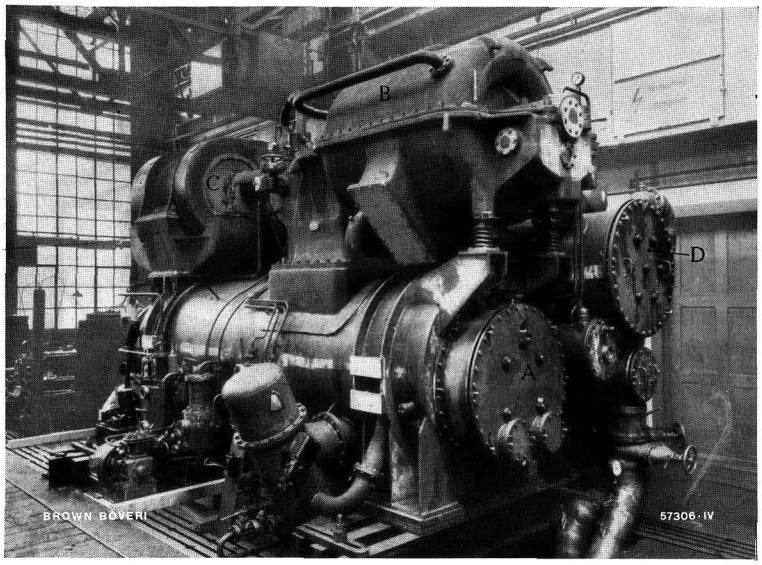


Fig. 21. — One of the two thermoblocs (heating machines) for the centralized heating and power plant of the Swiss Federal Institute of Technology, Zurich, on the test bed at Baden.

The two thermoblocs supplied to the Swiss Federal Institute of Technology at Zurich enable approximately 2600 tons of fuel to be saved yearly with pre-war operating conditions.

A. Evaporator in which the working medium (freon 11) boils and thus absorbs heat from the river water circulating through it.
B. Turbo-compressor.
C. Synchronous motor.
D. Condenser in which the evaporated and compressed working medium is condensed by the heating water, to which it gives up heat.

Heating output . . Mio kcal/h	1·5	2·0	2·5
Limmat water temperature on entering evaporator of heat pump °C	2	9	16
Limmat water quantity . m^3/min	1200	1200	1200
Hot-water leaving temperature °C	68	71	74
Hot-water return temperature °C	53	45	37
Hot-water quantity . . . m^3/h	100	77	68
Power input at the motor terminals with unity power factor . kW	610	775	960

The turbo-compressor is driven directly by a two-pole synchronous motor, having a continuous rating of 1100 kVA.

As can be seen in Fig. 22[1], the entire installation is underground.

The centralized heating plant of the Swiss Federal Institute of Technology supplies at the present time (Fig. 23) the needs of 17 buildings belonging to this establishment, 37 buildings belonging to the Zurich Cantonal Hospital, 3 large buildings of the Zurich Cantonal Administration and 85 private buildings. The total connected-up heating capacity amounts to roughly 38 Mio kcal/h, to which corresponds in a normal year a heat production of forty milliard kcal per year. The heat was produced hitherto by coal-fired boilers, one oil-fired Velox steam generator, as well as an electric boiler[2].

The addition of a heat-pump plant to the existing centralized-heating power station was made with the object of increasing its efficiency and saving fuel. The consumers purchase heat in the form of steam, heating water and hot water for domestic uses. Fig. 24 shows how the heat pumps will be connected up to the centralized heating system; they will supply heat in the first place to buildings situated close to the Limmat. The remaining heat is transmitted by means of a compensating pipe line to the main station, whence it is distributed to the consumers.

VI. THE HEAT-PUMP FOR HEATING AND COOLING IN AIR-CONDITIONING PLANTS.

The purpose of air conditioning is to establish and maintain air in closed rooms at a given state of temperature, humidity, cleanliness, movement, etc., regardless of the outdoor conditions. At its inception, only optimum conditions of well-being (comfort zone) were desired for people in cinemas, theatres, offices and factories, etc., but at the present time its scope has been extended to manufacturing and technological processes. The purpose, however, remains the same, and invariably consists of maintaining constant within a narrow margin both the humidity and temperature of the air (Fig. 25). These limits are particularly fine for certain technological processes. Such conditions imply that as a rule, air has to be heated and humidified in winter, whereas it must be cooled and dried in summer. Air-conditioning plants require therefore appliances for both cooling and heating. Since these two requirements together can be met by a heat pump, these machines are particularly well suited for plants of this description, the double use ensuring a comparatively high number of working hours per year.

An instance of a heat pump for air conditioning is afforded by the plant at the Congress Building in Zurich, which has already been alluded to on page 154.

[1] Particulars for Figs. 22—24 were kindly placed at our disposal by the administration of the centralized heating plant of the Swiss Federal Institute of Technology, Zurich.

[2] See: "Das Maschinenlaboratorium der Eidg. Techn. Hochschule, Zurich", reprint from the Schweizer Bauzeitung, Zurich, 1934/39.

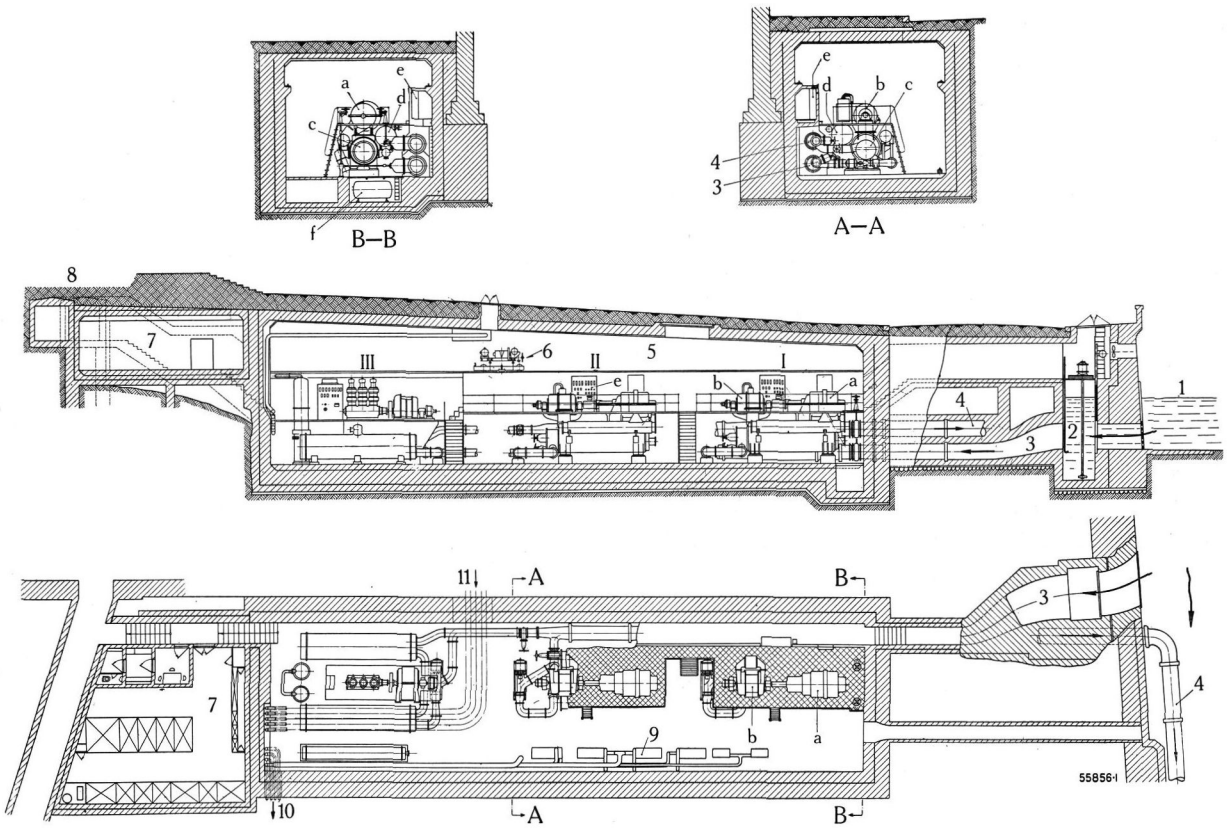


Fig. 22. — Subterranean heat pump station of the centralized heating and power plant of the Swiss Federal Institute of Technology, Zurich.

I, II. Brown Boveri thermoblocs. III. Third heat pump unit with reciprocating compressor.

1. River Limmat (source of heat).
2. Water intake.
3. Intake pipeline.
4. Water outlet pipeline.
5. Machine room with heat pumps I, II and III.
6. Machine-room crane.
7. High-tension room.
8. Entrance.
9. Hot-water circulating pumps.
10. Leaving pipelines for hot water.
11. Return pipelines for hot water.

a. Turbo-compressor.
b. Driving motor.
c. Evaporator.
d. Condenser.
e. Switchboard.
f. Freon-11 container.

Electrically-driven heat pump plants are particularly well suited for thickly-populated districts, as they require no boiler houses with their attendant chimneys, smoke and grime.

It comprises a Lèbre heat pump, laid out by the heat-pump research committee of the Swiss Federal Institute of Technology, Brown Boveri having supplied part of the equipment. It has an output of 50,000 kcal/h when heating and 27,000 kcal/h when cooling[1].

The main object of air conditioning in industrial plants is to maintain the most favourable conditions for the workers and maximum output, and particularly to improve the quality of the products.

Gear cutting in our works, for example, is carried out in a shop where the air is conditioned, and into which no daylight can penetrate[2], because it is only in this way that the requisite accuracy can be obtained. Similar conditions exist in plants manufacturing precision optical and measuring instruments.

Fig. 26 shows a projected installation for an optical-instrument works. The room temperature has to be maintained constantly at 22° C throughout the year, which implies, on the average, heating for 300 days and cooling for 65 days per year. The amount of fresh air taken in is only sufficient for its renewal. Part of the air coming from the rooms is re-utilized, and is mixed with the fresh air, thus going through a partly-closed circuit. This arrangement offers the advantage of reducing the temperature drop between incoming and outgoing air, and thus ensuring a more even distribution of heat. It may be recalled at this juncture that the smaller the temperature difference which has to be surmounted by the heat pump, the greater becomes its performance coefficient and therefore its efficiency.

[1] Bauer and Bolomey: "The Generation of Heat and Cold in Air Conditioning Plants by means of an electrically-driven Heat Pump" Elektrizitätsverwertung, 1939/40, No. 9/10.

[2] A. Meldahl: "Why does a gear sing? How can singing be avoided?" The Brown Boveri Review, No. 9/10, 1942, p. 284—289.

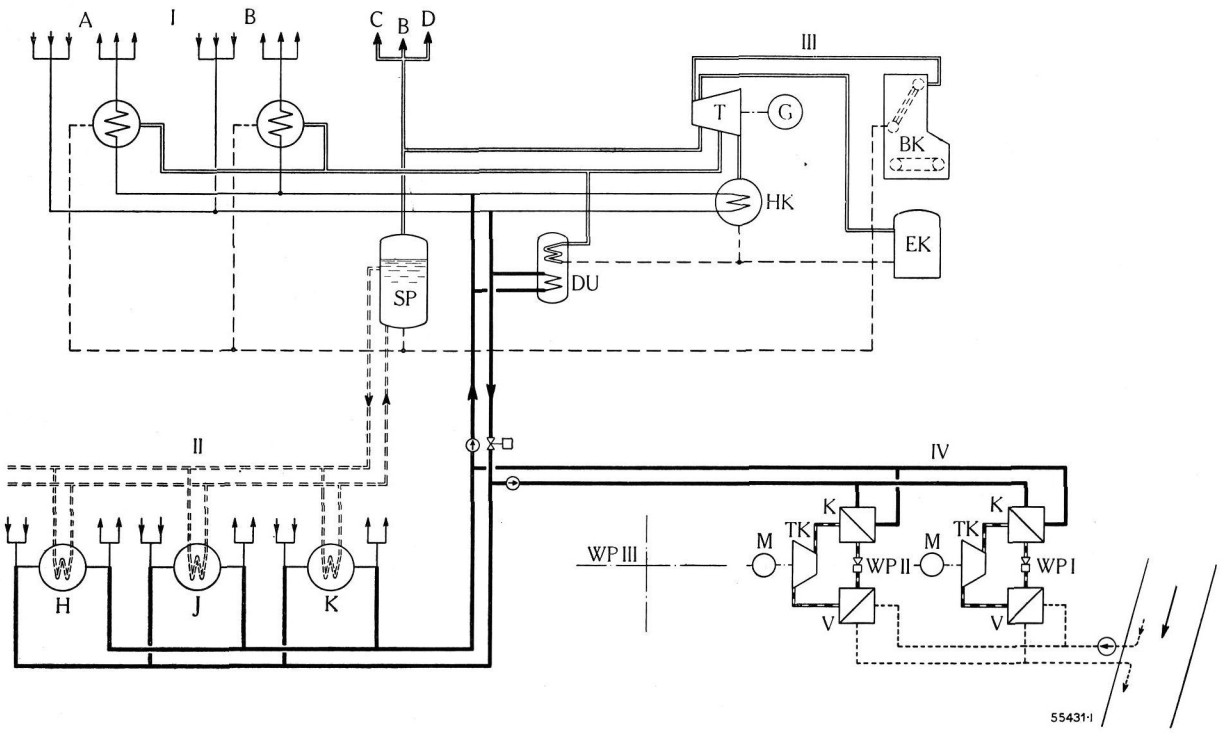


Fig. 23. — Layout diagram of the centralized heating and power plant of the Swiss Federal Institute of Technology, Zurich, with the additional heat-pump station.

The existing centralized heating and power plant with boilers heated by fuel and electricity is being supplemented by a heat pump plant, comprising three heating machines of which two are Brown Boveri thermoblocs, each having a heating capacity of 1·5—2·5 Mio kcal/h. This is the first large centralized heating plant in the world with heat pumps.

I. Consumers connected to main plant.
- A. Private dwellings.
- B. Swiss Federal Institute of Technology.
- C. Women's hospital.
- D. Cantonal hospital.

II. Consumers near to the River Limmat.
- H. Limmatblick building.
- J. Cantonal Walche buildings.
- K. Caspar-Escher house.

III. Heat-producing equipment of main plant.
- T. Bleeder back-pressure turbine.
- G. Generator.
- HK. Heating condenser.
- DU. Steam-water heat exchanger.
- SP. Heat accumulator.
- BK. Fuel-fired boiler.
- EK. Electric boiler.

IV. Additional heat-pump station.
- WP I, WP II, WP III. Heat pump sets.
- V. Evaporator.
- K. Condenser.
- TK. Turbo-compressor.
- M. Motor.

——— Heating water.
– – – – – Limmat water.
=■=■= Circuit of working medium of heat pump.

═══ Steam.
——— Heating water.
==== High-pressure heating water for covering largest load peaks.
– – – – – Condensate.

The automatic regulation of the state of the air calls for continuous adjustment of the amount of heat furnished or removed by the air-conditioning plant, as well as of the humidity content of the air. Such a plant requires therefore very comprehensive devices for circulating the air (ventilation), drying, dehumidifying and precipitating fog, as well as humidifying, elimination of harmful gases or dust, cooling and heating, the control of which calls for carefully-studied regulation. The choice of the means adopted for heating, cooling, etc. depends on the process considered and on the degree of accuracy within which the conditions have to be maintained (base load or exact heating, cooling, etc.). They can vary from summer to winter. For fuller details, special treatises on this subject should be consulted[1].

Examples of industrial applications of air-conditioning plants with heat pumps can be easily multiplied, but only the following will be mentioned here:—

In spinning mills, ample humidification of the air increases the elasticity and strength of the fibres and renders them easier to manipulate.

In tobacco factories, the aroma is preserved by maintaining constantly the requisite degree of humidity. The improvement of the conditions under which to-

[1] E.g., M. Hottinger: "Lüftung in Klimaanlagen", publishers J. Springer, Berlin, 1940.

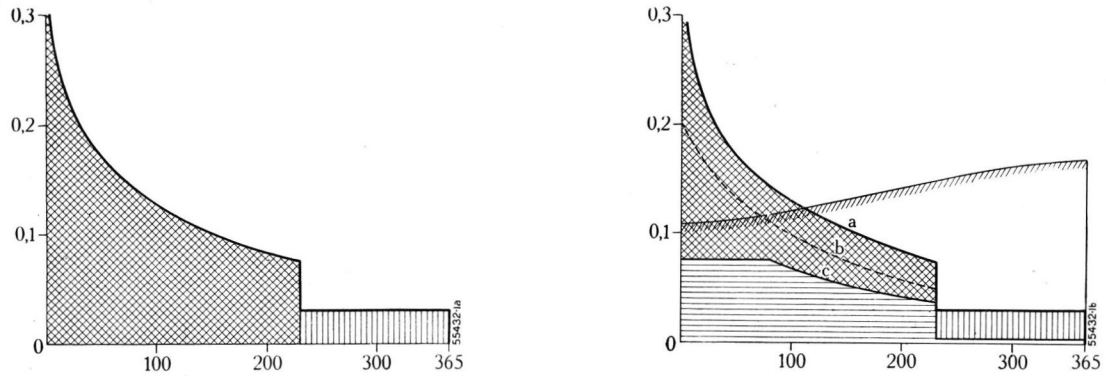


Fig. 24. — Diagram showing heating load taken over by the additional heat pump station of the centralized heating and power plant of the Swiss Federal Institute of Technology, Zurich.

Ordinates: Daily heat demand in 10^9 kcal.
Abscissae: Number of days per year.
Left: Diagram showing heat production before setting the additional heat-pump station to work.
Right: Diagram showing heat production after setting the additional heat-pump station to work.

Heat produced by coal-fired boilers.
Heat produced by electric boiler.
Heat produced by heat pumps.
Maximum amount of heat that can be produced by heat pumps.

The surface between curves a and b corresponds to the amount of heat consumed as steam (hospitals) which has to be produced as hitherto by boilers. The remaining surface between b and c is the amount of heat which with the maximum capacity of the heat pumps coming into consideration, still has to be obtained as hot water heated by the boilers at the main centralized heating and power station. It will be noted that the heat pump will enable considerable extensions to be made to the heating system.

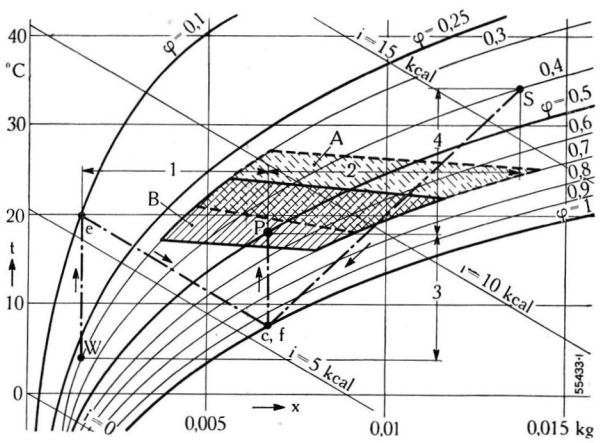


Fig. 25. — i—x diagram showing the conditioning process of air at atmospheric pressure.

Heat pumps are particularly well suited for air-conditioning plants, as they can be used for cooling in summer and heating in winter.

Air conditioning processes (heating and humidifying in summer, cooling and drying in summer) can be shown in a very clear manner with the aid of the i—x diagram*).

x. Water vapour content of a mixture comprising 1 kg of dry air and x kg of water vapour.

i. Total heat content of this mixture in kcal/kg.

φ. Relative humidity for the pressure $p_0 = 1$ kg/cm² absolute. $\varphi = 1$ = saturation (dew) line.

t. Air temperature.

A. Comfort zone for climatic conditions in Northern and Eastern Switzerland in summer.
$t = 27{\cdot}5 - 25°$ C and $21 - 18°$ C.
$\varphi = 0{\cdot}3 - 0{\cdot}7$.

B. Comfort zone for climatic conditions in Northern and Eastern Switzerland in winter.
$t = 24 - 22°$ C and $17 - 16°$ C.

P. Operating point corresponding to desired climatic conditions.
The modifications of state necessary to obtain P are as follows:
1. Humidifying.
2. Drying.
3. Heating.
4. Cooling.

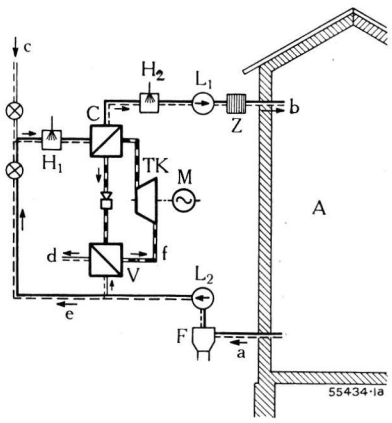
Example.

S. **State of outdoor air in summer.**
In order to reach the state corresponding to P, the air has to be cooled and at the same time moisture removed (S to c) with subsequent heating (c to P).

W. **State of outdoor air in winter.**
P is reached by heating (W to e), adiabatic humidifying (e to f) and further heating (f to P).

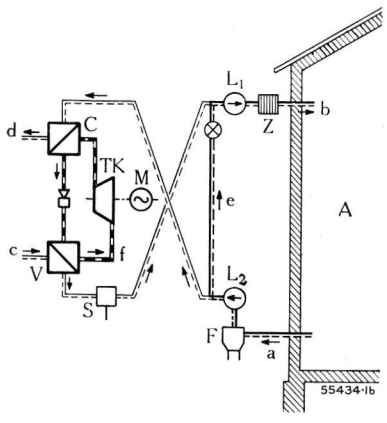
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*) For particulars regarding an extended i—x diagram for air, See: Ad. Baumann "Cooling and Drying Air (Air Conditioning) with particular Reference to the Air Expansion Process". The Brown Boveri Review 1930, p. 271.



a.

Winter connection with heat pump as heating machine.



b.

Summer connection with heat pump as cooling machine.

Figs. 26a and b. — Projected heat pump installation for an optical-instrument factory.

Conditioning air in workshops increases the output and accuracy, i. e., quality of the manufactured products.

A. Closed workshops.
M. Driving motor.
TK. Turbo-compressor
V. Evaporator
C. Condenser
} of the heat pump.
L_1. Fresh-air fan.
L_2. Exhaust-air fan.
D. Dust filter.
S. Water separator.
H_1, H_2. Humidifiers.
Z. Additional heating.
a. Outlet duct for exhaust air.
b. Inlet duct for conditioned fresh air.
c. Intake of fresh, outdoor air.
d. Part of exhaust air returned to the atmosphere.
e. Air going through a closed circuit, which is mixed with the fresh, outdoor air.
f. Working-medium circuit of the heat pump.

bacco is handled reduces wastage and improves both output and quality of the products.

In paper mills, printing and photographic works, a certain degree of humidity is necessary for preventing superficial changes of the products in question.

VII. EVAPORATING PLANTS WITH HEAT PUMPS (THERMO-COMPRESSORS).

Very great possibilities for utilizing heat pumps (thermo-compressors) exist in evaporating plants. The chemical industry offers a vast field in this connection, as, for example, manufacturing pharmaceutical products, concentrating dyes, evaporating solutions and mixtures of two or several substances, etc. The preparation of foodstuffs also calls for evaporating plants for concentrating unfermented fruit juices, condensed or powdered milk, whey for lactose (milk sugar), etc. Evaporating plants are also used extensively for the preparation of table salt and sugar. Finally, it is possible to obtain drinking water from sea and other water unfit for consumption by distillation with the aid of a thermo-compressor.

Steam compressors can be also used as rotary pressure changers, by increasing the pressure of expanded steam to a higher value suitable for a working process.

Conditions in plants of this description are very much more favourable for heat pumps than heating installations, for instance. The temperature differences which have to be surmounted are small, thus ensuring high performance coefficients, and, in addition, the plants themselves are, as has already been mentioned[1] very simple and cheap. The cost of evaporating a ton of water with a thermo-compressor amounts to only a fraction of the expenditure necessary to obtain the same result by heating with live steam. Evaporating plants heated with live steam coming from coal-fired boilers can be therefore replaced with advantage by installations employing electrically-driven thermo-compressors.

The evaporation of solutions calls for considerable quantities of heat, which, for instance, amount to 550 to 600 kcal per kilogram of water evaporated, according to the pressure and preheating, insofar as the latent heat of the steam given off is not recuperated. This condition can be met in certain cases by multiple-effect evaporators, which call, however, for a large expenditure of material. It can be fulfilled in a much more simple manner with the aid of a thermo-compressor, which operates with a high efficiency, since *a consumption of energy of one kWh permits of evaporating up to 10 to 35 kg of water*, according

[1] A. Meldahl: "The Heat Pump as a Refrigerating and Heating Machine", The Brown Boveri Review, Nos. 5/6, 1943, p. 75—85.

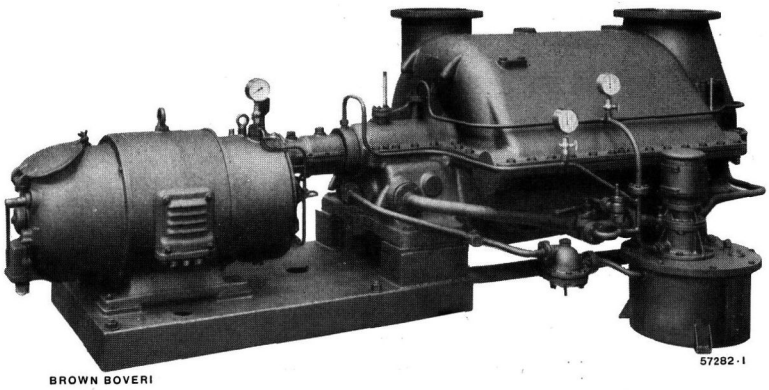


Fig. 27. — Thermo-compressor for handling a mixture of alcohol and water vapour.

(vapour pressure exchanger).

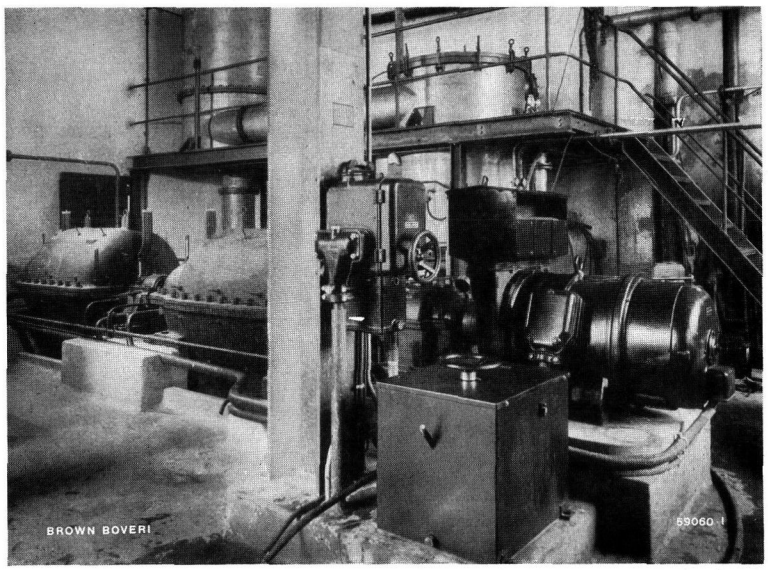
Weight of vapour handled	400 kg/h.
Suction pressure	0·0204 kg/cm² abs.
Delivery pressure	0·102 kg/cm² abs.

The gear built on the motor casing gives a very compact set.

The high vacuum inside the blower calls for special care and experience in designing the glands where the shaft traverses the casing.

Fig. 28. — Thermo-compressor in a chemical works for evaporating various solutions which are hard to boil.

The large pressure ratio rendered it necessary to provide a two-cylinder machine. The evaporation of 700 kg of water per hour at a pressure of 0·0606 kg/cm² absolute requires theoretically an expenditure of heat of 428,000 kcal or its equivalent of 500 kWh if heating is carried out directly. For the conditions coming into consideration here, roughly 5·3 times less electrical energy — i. e. 94 kWh — is required, thanks to the use of a thermo-compressor which enables the latent heat of the steam given off to be recuperated and utilized for heating the boiling solution. Thermo-compressors *enable an appreciable saving of electrical energy to be* obtained, which constitutes a very important asset at the present time in view of the general shortage of this commodity. The specific evaporation figure — i. e. the number of kilograms of water evaporated per kWh expended — amounts to 7·5 kg/kWh.



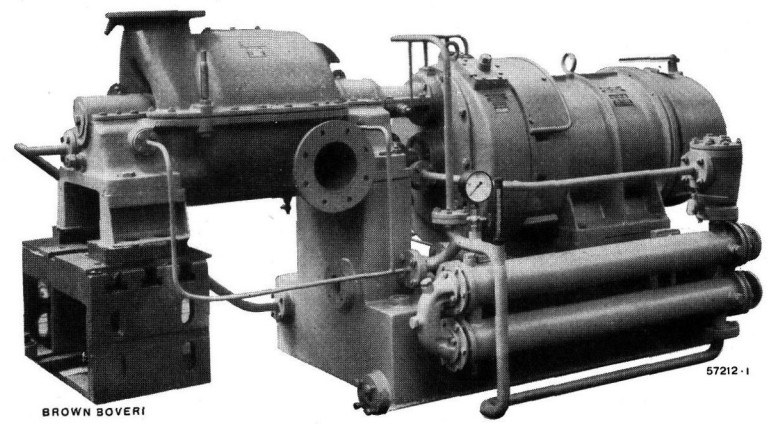


Fig. 29. — Thermo-compressor for a water distilling plant.

Weight of water evaporated	1250 kg/h.
Suction pressure	1·03 kg/cm² abs.
Corresponding boiling temperature	100° C.

Thanks to their compact design and small energy consumption distilling plants with thermo-compressors are suitable for both stationary and particularly mobile plants for distilling water where inadequate supplies are available for drinking.

This plant differs from that shown in Fig. 28 in that evaporation is carried out at a higher pressure, viz. *that of the atmosphere. This, together with the fact* that the compressor has to produce an appreciably lower pressure ratio, is the reason why the thermo-compressor set, notwithstanding its greater evaporative capacity, is considerably smaller than that shown in Fig. 28.

to the operating pressures, nature and concentration of the substance treated.

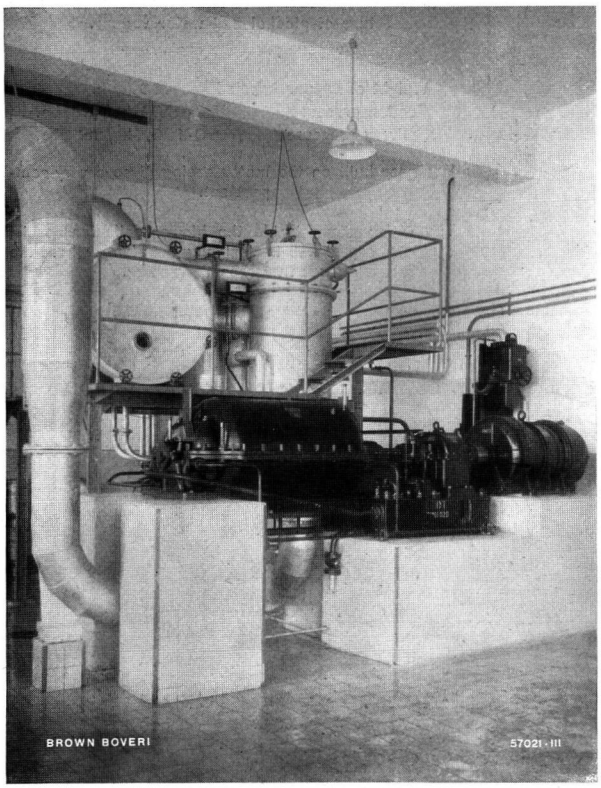


Fig. 30. — Thermo-compressor in an evaporating plant handling milk products.

Nominal evaporative capacity: 1000 kg of water per hour.

Evaporation with thermo-compressors affords an ideal means of concentrating liquid foodstuffs. Among the many advantages they possess may be mentioned:

a. Concentration can be carried out at the moderate temperatures necessary for conserving vitamines, the flavour and aroma.

b. The various operations are carried out with maximum cleanliness and hygiene due to the use of electricity and the suppression of fuel.

c. Due to the high performance coefficient, the energy consumption amounts to only a small fraction of that necessary for evaporation by direct electric heating.

The ratio $\lambda = \frac{G}{L}$ is called specific evaporation figure, G being the amount of water evaporated in kg per hour and L the power input measured at the terminals of the motor driving the thermo-compressor.

Both performance coefficient ε and specific evaporation figure become higher as the temperature level — i.e., the operating pressures — is raised and the temperature difference Δt between the saturation temperatures corresponding to the pressures at the outlet and inlet of the thermo-compressor become small.

Example showing the influence of the pressure level.

Weight of water to be evaporated 1000 kg/h. Boiling temperature of the solution of $t_0 = 45^0$ C, $T_0 = 318^0$ K and a temperature difference $\Delta t = 14^0$ C, hence $t_1 = 59^0$ C and $T_1 = 332^0$ K. With the given size of plant, an overall efficiency of $\eta = 0{\cdot}55$ can be assumed.

The performance coefficient amounts to

$$\varepsilon = \frac{332}{14} \times 0{\cdot}55 = 13{\cdot}05$$

The power input at the motor terminals can be determined from this figure as being equal to 58 kW. Hence:

$$\lambda = \frac{1000}{58} = 17{\cdot}2 \text{ kg/kWh.}$$

If, on the other hand, evaporation is carried out under atmospheric pressure, i. e., $t_0 = 99^0$ C, $T_0 = 372^0$ K, and all other conditions remaining the same, so that $t_1 = 113^0$, $T_1 = 386^0$ K, then:

$$\varepsilon = \frac{386}{14} \cdot 0{\cdot}55 = 15{\cdot}10 \text{ and}$$

$$\lambda = \frac{1000}{44{\cdot}5} = 22{\cdot}5 \text{ kg/kWh}$$

with a power input of 44·5 kW at the motor terminals.

Evaporation is carried out preferably, whenever possible, at atmospheric pressure, because no special sealing devices are then required for the compressor glands, nor is a special pump required for evacuating condensed water, which simplifies the operation of the plant.

Certain substances, however, are affected by the temperature, so that their concentration has to be carried out at low temperatures, under vacuum, so as to maintain their vitamin content, flavour and aroma. The treatment of milk, fruit juice and certain organic solutions calls for precautions of this description.

The power input of the compressor should correspond exactly to the amount of heat necessary to cover the losses of the evaporating plant, such as radiation, warming up the raw solution, heat carried away with the concentrated product and condensate, heat removed by the vacuum pump, etc. If the compressor has been correctly dimensioned, both heat and temperature balance should tally.

The difference in saturation temperatures between compressed steam and boiling solution determines the pressure ratio which has to be produced by the compressor, and consequently the number of impellers. The effective temperature difference, i. e., pressure ratio, is however considerably higher than that necessary for heat transmission across the heating surfaces alone. Account must be also taken of the pressure losses of the apparatus and pipework, air leakage and, particu-

larly if high vacua are being worked to, of the height of the column of boiling liquid, as well as of the formation of deposits on the heating surfaces and the lag of the boiling point. This last-named factor is the difference in temperature of the boiling point of the solution and that of pure water, and generally increases as concentration progresses.

An exact knowledge of all these factors is necessary for each particular case in order to ensure satisfactory operation under all conditions occurring.

in Figs. 27 to 31. The installation shown in Fig. 30 is especially interesting[1]; it is used for preparing milk sugar (lactose), as a by-product of casein and albumine. Full or skimmed milk can be also concentrated in the initial stages of manufacturing dried milk products, which are very stable and can be kept for a considerable time. In this manner, seasonal surpluses can be stored for times when there is a shortage of milk. Such plants can operate entirely without fuel, and can be paid off within a very short time.

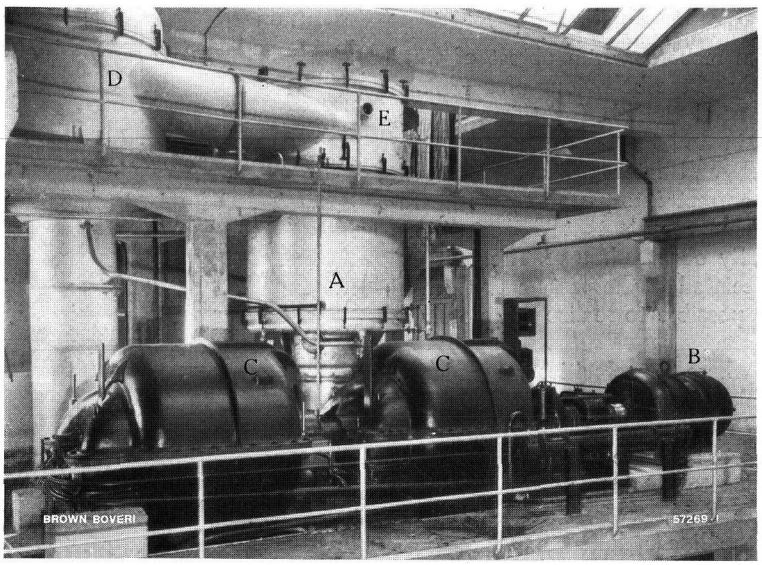


Fig. 31. — Thermo-compressor set for an evaporating plant handling milk products and unfermented fruit juice, 3000 kg of water being normally evaporated per hour.

The liquid to be concentrated boils in the evaporator A. The compressor C, driven by an induction motor B over gearing, takes in the water vapour which is boiled off after it has previously traversed the steam separator D, and delivers it to the heating tubes of the evaporator A, where its latent heat is used for heating. Inspection windows E at appropriate places enable the operation to be readily checked.

The energy consumption is approximately 9 times smaller than with direct electric heating, a saving which enables the plant to be rapidly paid off.

According to the available raw products, the installation is used variously and without any changes, for concentrating either milk products or unfermented fruit juices.

The properties of the liquid to be treated also call for particular attention when designing the compressor. Suitable separators, dryers, washers, etc., prevent incrustation of the inner surfaces, whereas corrosion of the compressor blading can be avoided by an appropriate choice of the materials used. With plants working under vacuum special arrangements for evacuating the system and preventing ingress of air are required.

Whenever different kinds of products are treated in the same plant, different temperature differences may occur. Various means exist for adapting the compressor to the changed operating conditions, among which speed regulation may be mentioned.

Some representative examples of thermo-compressors for evaporating or distilling plants are shown

All kinds of products can be treated, like whey and milk sugar as well as milk powder and dried milk in the spring, whereas various kinds of fruit juices can be concentrated later on in the year. A plant of this description was set to work in Eastern Switzerland in autumn 1942 for evaporating 3000 kg of water per hour (Fig. 31). During the autumn, up to 30,000 litres of grape juice were treated daily. It is worth mentioning that this plant was manufactured and set to work in the very short time of four months.

(MS 981)
(BBR 7/8 43)

Ad. Baumann and D. Marples.

[1] See Ad. Baumann (Dept. V): "The Thermo-compressor Installation in Lucens for the Concentration of Milk", The Brown Boveri Review, No. 5/6, 1943, page 85.

SOME CONSIDERATIONS INFLUENCING THE DESIGN OF WIND TUNNELS.

Decimal Index 533.6.07

We are pleased to be able to publish in this number a study by Mr. G. Darrieus, of our French licencee, the Cie. Electro-Mécanique of Paris, on the design of wind tunnels.

In this article, Mr. Darrieus, who is acknowledged to be an authority on the subject, deals with the characteristics and the design of the main components, comparing in particular open wind tunnels with those operating on the return-flow principle, and discusses the relative advantages of test sections with guided and with free jets.

I. CLASSIFICATION ON CHOICE OF TYPE OF WIND TUNNEL.

THE oldest, and the most common object of wind tunnels is the study of the behaviour of aeroplane models. The use of wind tunnels for this purpose, which was suggested in 1908 almost simultaneously by Rateau and Eiffel, and later again by Prandtl, necessitates, for the correct interpretation of the results obtained with scale models that the conditions for similitude be strictly observed, a requirement not always properly appreciated in the past, and which in fact it is not always possible to realize completely in practice.

Thus, for instance, where the velocities are moderate compared with the velocity of sound, the observance of the essential condition for similitude of the same ratio of the inertia forces to the forces of viscosity, would require, in order to obtain the same Reynold's number with the same fluid, that the speed be increased inversely as the linear dimensions.

There is, however, an obstacle to the realization of this condition. Due to the high velocities (comparable with those of sound) thus rendered necessary, the compressibility of the medium begins to become manifest, requiring the use of a second criterion (Mach, Sarrau) involving the ratio of the velocity of air to the velocity of sound, the maintenance of which is incompatible with the maintenance of the same Reynold's number.

Moreover, these high velocities would necessitate very high blower powers, so that apart from the impractibility of submitting the model to the large forces which it would then have to withstand, the object of the wind tunnel, namely, the reduction of the experimental costs, would not be attained.

The only solution enabling the maintenance of the same Reynold's number without increasing the velocity, consists in increasing the density of the air in inverse ratio to the reduction of the linear dimensions, thus leading to the creation of variable density wind tunnels. Such tunnels are, unfortunately very expensive to build, so that there are only a few in existence, namely, one for 20 kg/cm^2 abs at Langley Field in the U. S. A., and one for 25 kg/cm^2 abs at Teddington in England.

Thus, for obvious reasons of cost, and because of the rapid progress in flying speeds, the velocities in wind tunnels are almost always inferior to those reached in practice, in spite of the fact that out of regard for the maintenance of the Mach number, the attainment of such speeds would appear to become more and more desirable as flight velocities approach those of sound.

Fortunately, experience shows that, particularly for the usually well stream-lined shapes of aeroplane bodies and wings, the sacrifice of the Reynold's number does not, in general, entail any serious consequences in the application of the principle of similitude and in the extension of the results obtained with scale models to the full size object, which can be carried out with a sufficiently high degree of accuracy, by the judicious use of scale effect corrections involving the separation of the resistances (induced, form, profile, etc.) as was done with complete success by Froude in his tank tests on ships' hulls.

The admission of this frequent necessity of accepting compromises, the possible vitiating effects of which must always be remembered, underlines the importance of the application of sound scientific reasoning, experience, keen discernment and plain common sense, to the correct interpretation of the results obtained with small scale models.

It must, however, be borne in mind that, although this method of investigation, which for a long while was regarded with suspicion, has, because of its notable successes now achieved great popularity, so that there is a tendency in some parts to look upon it as a panacea, the qualities which it demands of those who may be expected to apply it with success are of a more than usual nature, and must not be under-estimated.

Another important condition, often overlooked, for the fidelity of the tests made on models, is that the jet of air in which they are immersed shall be at least sufficiently large compared with the dimensions of the model, particularly in the transverse direction, in order that the more or less exact theoretical corrections, made necessary by the boundary conditions shall remain within moderate limits.

Generally, because of the better accessibility of the model, an open jet, that is one traversing the test chamber without any lateral guidance, is preferred to a closed one; but this greater convenience for the mounting of the model is obtained at the cost of a relatively large pressure drop and of an appreciable amount of turbulence in the transition layer, separating the jet containing the model from the more or less stagnant atmosphere of the room.

The corresponding loss, referred to the energy per unit mass $\frac{\rho}{2} w^2$ in the jet, is obtained simply by multiplying the average coefficient of friction $c_f = 0 \cdot 02$ along this surface of discontinuity, by the ratio of the area of this latter to the section of the jet, i. e., by $4 \frac{l}{d}$, where l and d denote the length and diameter



Fig. 1. — Large open wind tunnel with free jet for testing engines and aeroplane bodies.
View looking at suction inlet.
Jet diameter 5 m. Length of free part of jet 8 m. Maximum speed 103 m/s.
Power absorbed by fan 3800 kW.

respectively, of the free portion of the jet. This loss reaches already 12 % for a length between the nozzle and the diffusor equal to 1·5 diameters, whereas the corresponding loss for a fully guided jet, which depends on the very much smaller coefficient of friction (of the order of 0·002 for the Reynold's numbers here encountered) between the fluid and the solid wall, would be about ten times less.

Since the efficiency or coefficient of use of the tunnel[1] depends to a large extent upon this loss, it would seem advisable, out of consideration of the power required to obtain a particular result, in spite of the disadvantage of poorer accessibility, to employ a guided jet, which due to its 1·5—2 times higher coefficient of use would enable a very much larger jet and especially a very much greater homogeneous core to be obtained with the same power.

An interesting compromise which has been proposed on a number of occasions, but which does not appear ever to have been put into practice in spite of the fact that it offers definite advantages, consists in the use of a partially guided jet, that is one guided only by a roof and a floor connecting the mouth of the nozzle with the diffusor, and leaving large openings on each side of the jet, thus affording almost the same accessibility as an open jet.

Such a solution offers in addition to the increase of efficiency resulting from the reduced pressure loss during the passage of the jet through the test section, the further advantage that the corrections for taking into account the finite dimensions of the jet, either vanish or become very small, as might be expected from the fact that the corrections for open and closed jets are equal in magnitude, but opposite in sign.

II. COMPARISON BETWEEN OPEN AND RETURN-FLOW TYPES OF WIND TUNNELS.

The first wind tunnels built by Eiffel and practically all those erected in France since that time, are of the open type (Fig. 1); that is to say, the air is drawn in from the outside atmosphere or from the building containing the tunnel, where the return-flow velocity is negligible.

There is, therefore, a lowering of the pressure in the test chamber through which the jet passes, which is proportional to the square of the velocity of the jet, necessitating special arrangements for communication with the exterior during operation, and possibly interfering with physiological observations when the depression attains appreciable values (about 85 mm w.g. for 140 m/s).

The majority of modern wind tunnels are therefore of the return-flow type, an arrangement first employed by Prandtl, and in which the air flows in a closed circuit (Fig. 2), thus enabling the test chamber to be left in communication with the atmosphere at normal pressure.

Whereas with an open tunnel, the residual velocity of the air at the outlet of the diffusor is lost, the return-flow arrangement attempts to conserve this velocity and the corresponding kinetic energy, thus helping to overcome the pressure drop in the bends and along the ducts.

By employing deflecting vanes of a suitable profile, the pressure drop in the bends can be reduced to $0 \cdot 15 \frac{\rho}{2} w^2$ (ρ being the density and w the mean velocity of the air normal to the corresponding sections), but even with this precaution the four changes of direction and the straight paths separating them would give rise to intolerably high losses if the velocity were not as in the case of open type tunnels progressively reduced along the circuit by a series of successive enlargements of the cross-section.

Such a measure out of regard for the efficiency is, however, in line with the more recent preoccupation

[1] The coefficient of use q or quality of the wind tunnel (including the blower) is the ratio of the isentropic power L_0 of the compressor which would be necessary without any recuperation of the velocity, to the power L, actually taken by the compressor or,

$$q = L_0 : L = \int_{p-\delta p}^{p} \frac{dp}{\rho} : L$$

where p is the pressure in front of the nozzle, $p-\delta p$ the pressure in the test chamber, and ρ the density along an isotropic line, passing through a point corresponding to the conditions at the compressor suction. For moderate Mach numbers, $q = \eta \, \Delta p_{nozzle} : \Delta p$ compressor, η being the efficiency of the compressor, and $\Delta p_{nozzle} : \Delta p_{compressor}$ a factor, which for low Mach numbers corresponds to the loss coefficient of the circuit.

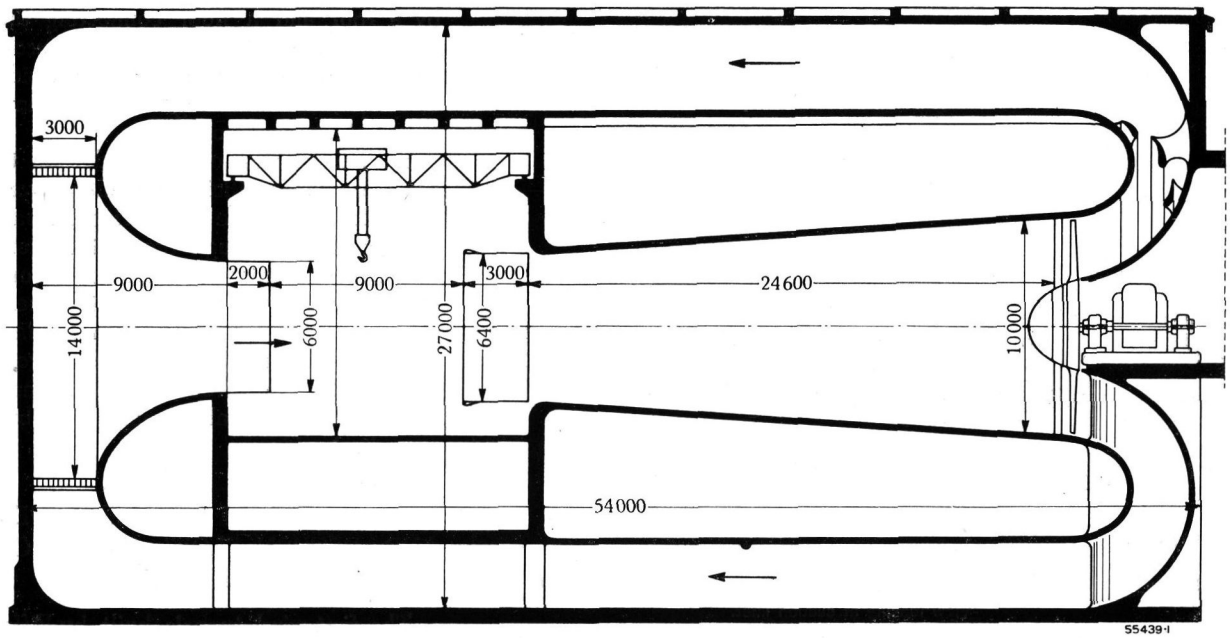


Fig. 2. — Return-flow wind tunnel.

The air flows from the diffusor outlet back to the inlet through a cylindrical stabilizing grid in front of the nozzle.

towards reducing the turbulence of the jet which also leads to the use of large sections in the return-flow section and of large air volumes where the air has time to settle before again entering the filter; so that in general, although the open tunnel of the Eiffel type is longer owing to the length of their diffusor, the return-flow type of tunnel, being greater in width, occupies finally an area at least as large.

The natural temptation to keep the costs low even at the expense of the efficiency may, however, lead to an excessive reduction in the size of large return-flow tunnels, so that in some cases it was found that the wake set up by the model did not have time to dissipate itself in the journey round the circuit, giving rise to troublesome interference or to uncertain corrections.

On the other hand, return-flow tunnels, unlike open tunnels, at any rate those drawing the air in from the atmosphere, are unaffected by the weather conditions or by the prevailing wind, but this advantage is partly nullified in the case of large installations, by the necessity of renewing a considerable fraction at every circuit due to the rapid heating up of the air.

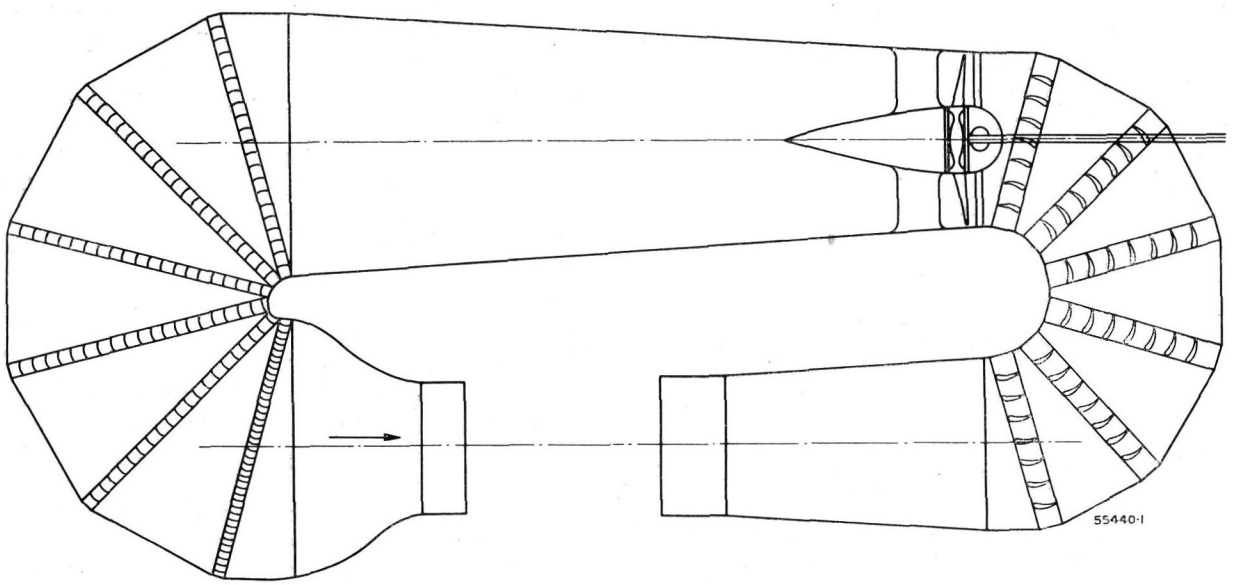


Fig. 3. — Return-flow wind tunnel.

With deflecting vanes at the bends split up into steps of 30°.

For instance, the rise of temperature resulting from the destruction of a velocity of 200 m/s amounts to $\frac{1}{2}\frac{(200)^2}{1000} = 20^0$ C*, the final increase in temperature per circuit will be 3^0 C assuming the quality figure of the tunnel to be as high as 7, so that a 10% renewal even if followed by a perfect mixing, would still leave a rise of temperature of 30^0 C, which in most cases could not be allowed.

The difficulty is further increased in the case of tests on engine gondolas with running engines, the exhaust gases of which not only contribute a rise in temperature of the same order of magnitude, but also rapidly vitiate the circulated air, so that for such tests, an open tunnel exhausting to atmosphere, is certainly preferable.

In regard to the efficiency or coefficient of use, which depends on many factors capable of variation at will between fairly large limits, it may be said that neither type, open or return-flow, presents any fundamental advantage over the other.

The loss coefficient of 0·15 applying to bends containing a grid of deflecting vanes, is very high, even for the best profiles, compared with the value which might be expected on the basis of tests on individual profiles. The reason lies undoubtedly in the magnitude of the deflection (90^0) which these vanes must impart to the stream and in the high value of the resulting tip losses.**

Since these losses, being of the same nature as induced resistance, are roughly proportional to the square of the circulation, it is to be expected as indeed has been proved by experiments on compressor blade rings, that they could be considerably reduced by splitting up the deflection into several stages of for instance 30^0 as illustrated in Fig. 3. The total surface of the vanes is not increased, and it is possible to increase slightly the section of each successive row to compensate at least as far as the static pressure is concerned, the loss due to the grating.

III. THE FILTER AND THE REDUCTION OF THE TURBULENCE IN THE JET.

If the air could be aspirated into the nozzle from an infinitely great, absolutely still air space, the law of conservation of irrotational motion would imply that the simple acceleration due to the pressure gradient would suffice to produce a perfectly homogeneous flow, devoid of all turbulence, except in the thin boundary layer at the converging walls of the nozzle. In reality, both with return-flow and with open tunnels drawing in air from a large building or from the atmosphere, where some wind is almost invariably present, there are always certain initial residual velocities, the transversal components of which are magnified by the suction effect of the nozzle into longitudinal eddies or vortices which, according to experience, occasionally manifest themselves by condensation of moisture, or by shadows cast by the refraction of incident light.

If we denote by n, the ratio in which the mean velocity u is magnified by the nozzle, or the corresponding ratio of the sectional areas beyond a certain point where

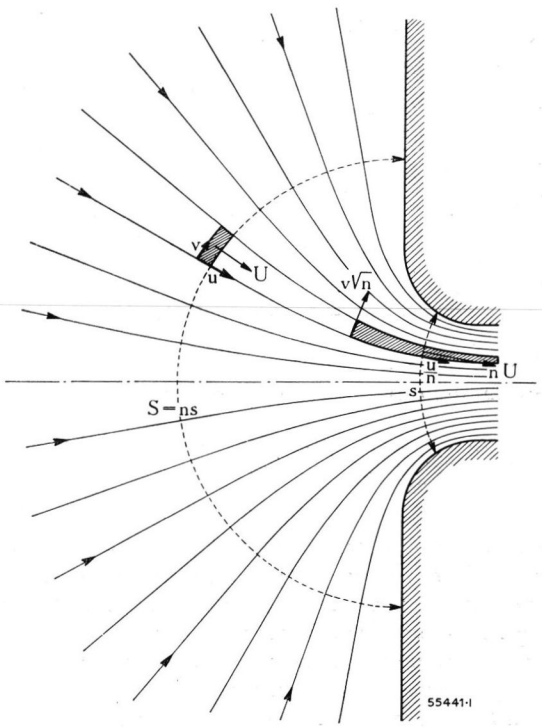


Fig. 4. — Conditions at the nozzle inlet.

As a consequence of the Helmholtz vortex law an increase of the velocity in the ratio of n causes the longitudinal components of the turbulence to diminish in the ratio of n, and the transversal components to increase in the ratio of $\sqrt{n}$.

the initial components of the stray velocities due to turbulence along and perpendicular to the direction of flow are u, v and w, respectively, then as shown by Prandtl, according to the Cauchy-Helmholtz theory of the conservation of the circulation around the boundary of the fluid, an elementary parallelepiped would be elongated in the ratio of n, and its transverse dimensions would be contracted in the ratio of $\sqrt{n}$, implying alterations in the inverse ratio of the fluctuating velocities which become $\frac{u}{n}$, $v\sqrt{n}$ and $w\sqrt{n}$, respectively. The fictitious pressures along the three principal axes ρu^2, ρv^2 and ρw^2 associated with the correlative exchanges of motion together with the corresponding contributions of the kinetic energy, are thus multiplied by the factors $\frac{1}{n^2}$, n and n.

If, for instance, the initial turbulence is isotropic, i. e., the mean values of the velocity components along the three axis are equal ($u^2 = v^2 = w^2$), the final energy will be increased in the ratio $\frac{1}{3}\left(\frac{1}{n^2} + 2n\right)$,

* The specific heat of air at constant pressure and at normal temperatures is very nearly 1000 kilojoules per ton ^{0}C.

** A. Meldahl, The Brown Boveri Review Nov., 1941, p. 356.

that is, 6·6 if $n = 10$, so that, although the rapid contraction of the flow tends to equalize the axial velocity by reducing its fluctuations in the absolute ratio of $\frac{1}{n^2}$ or in the relative ratio of $\frac{1}{n^2}$ and diminishes in the ratio of $\frac{\sqrt{n}}{n} = \frac{1}{\sqrt{n}}$ the angle of deviation of the stream lines, it introduces a considerable increase in the energy of turbulence. This increase of energy is, moreover, associated with an additional

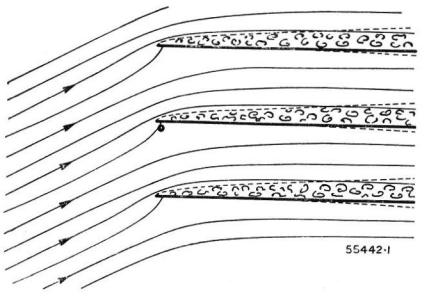


Fig. 5. — Stabilizing filter consisting of thin plates.

Due to the initial turbulence, the incident flow is not normal, resulting in separation and in an increase of the pressure drop.

pressure loss due to the increase of the transverse pressures ρv^2, ρw^2 by means of which the fluid so to speak supports itself against the converging walls of the nozzle.*

* The investigations of the author (5th Congress of Applied Mechanics at Cambridge, U. S. A., 1938) on the mechanics of the energy exchange between cellular vortices into which turbulence can be resolved, suggests the possibility of calculating a period of decay for the temporary anisotropy "adiabatically" introduced into the fluid by its sudden deformation, and in this way, starting from the experimentally determined turbulence spectrum, to arrive at a rational interpretation and determination of the length of the mixture path of Prandtl.

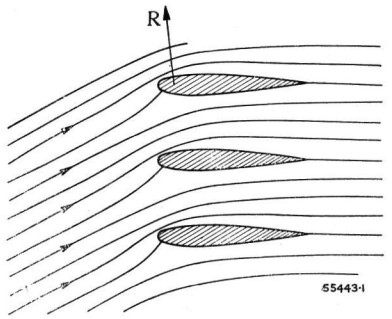


Fig. 6. — Stabilizing filter with stream-lined elements.

Separation no longer takes place when the incident flow is not normal, thus reducing the pressure drop. The reaction of the blades which is normal to the vector average of the inlet and outlet velocities causes a small increase in pressure corresponding to a partial recuperation of the turbulence energy.

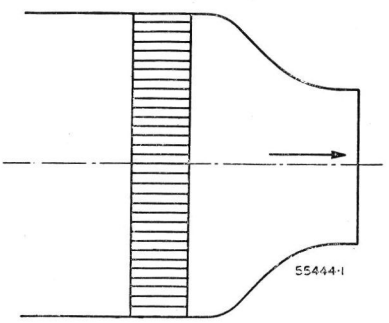


Fig. 7. — Stabilizing filter.

Common arrangement in a return-flow tunnel.

The best location for the filter or honeycomb destined to rectify the lines of flow and to render the jet uniform is, therefore, at the inlet to the nozzle, with dimensions as large as permitted by considerations of symmetry about the axis, because, not only is the corresponding pressure drop, which is proportional to the square of the velocity, thus

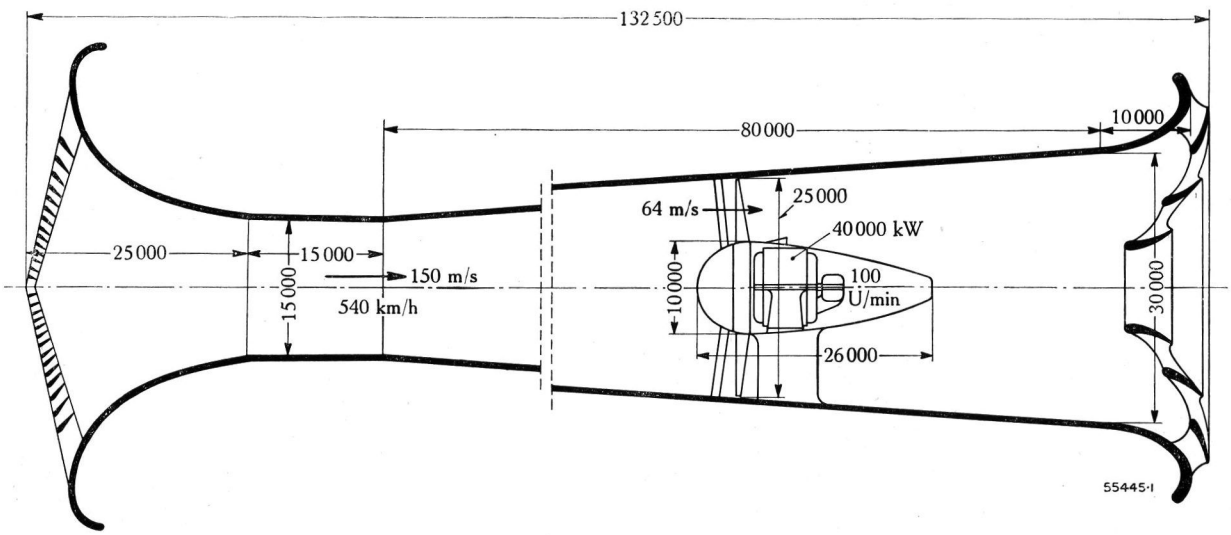


Fig. 8. — High-speed wind-tunnel.

With guided jet for tests on engine gondolas. Diameter of jet 15 m. Air speed 150 m/s (540 km/h). Motor power 40,000 kW.

reduced, but the filter is also much more effective due to the smaller amount of turbulence which has to be destroyed, the subsequent rectification of the stream lines in the converging portion, and finally, the partial dissipation of the fine residual turbulence caused by the wake of the vanes of the filter, in the longer path up to the test chamber.

It is customary, for the sake of simplicity and of economy, to build the filter of thin flat plates (Fig. 5), but stream-lined profiles with well-rounded noses are preferable, because the pressure drop with an initially uniform and parallel flow is no greater than that for flat plates (in which case, no filter is necessary), whilst the absence of separation in case of oblique incidence gives them a superiority which manifests itself in a considerable reduction of the wake, of the pressure drop and of their disturbing effects.

Still more advantageous is the fact that in a grid composed of ideal aerofoils, the reaction of the fluid is perpendicular to the vector average of the inlet and outlet velocities, which means that there is a component along the direction of flow which may even exceed the pressure drop due to friction, and representing a recuperation at first sight paradoxal, but nevertheless theoretically possible, at least as far as the main components are concerned, of the turbulence energy.

This phenomena is entirely analogous to that observed in quite old experiments on the spontaneous propulsion of ships by wave action acting on stream-lined ailerons rigidly attached to the hull or serving as anti-rolling keels, and which in troubled seas are subjected to a forward thrust.*

* La stabilisation propulsive des navires, by M. Le Besnerais, Association Technique Maritime et Aéronautique, 1936, p. 647.

The majority of filters in existing wind tunnel installations still consist of a few coarse pitch relatively short cells which only partly fulfil their role by substituting for the initial turbulence, the marked and persistent wake of their partitions, whereas the acknowledged importance of a jet as nearly free from turbulence as the atmospheric air at a sufficient distance from the ground, for a correct interpretation of the varied wind tunnel tests, particularly of those made on wing profiles, now leads to the use of closer pitched and generally longer filters, the residual turbulence of which, being much finer is much more rapidly dissipated. Experiments made with a temporary grid which for reasons of economy was placed near to the end section of a 5 cm diameter nozzle showed that although the level of the fine turbulence thus obtained was relatively low ($\sqrt{u^2}$: V less than 1 %) the individual wakes of the plates of 100 mm pitch and 200 mm depth, at higher velocities, and contrary to current views, could still be distinctly detected at distances up to 8 m from the mouth of the nozzle.

For return-flow wind tunnels, the simplest type of filter consisting of prismatic cells located at the widest section, is quite suitable. The depth of the cells may be as much as 5 to 10 times the pitch if it is desired to reach a very low turbulence level (in some recent installations the use of a very fine filter combined with large reduction ratio n of the section has enabled the ratio $\sqrt{u^2}$: V to be reduced to less than 1/1000). A particularly favourable form of filter for open tunnels (Fig. 8) respecting the symmetry of flow about the axis (which is not the case with filters of the converging square cell type) and which, not comprising any generated surface, lends itself readily to

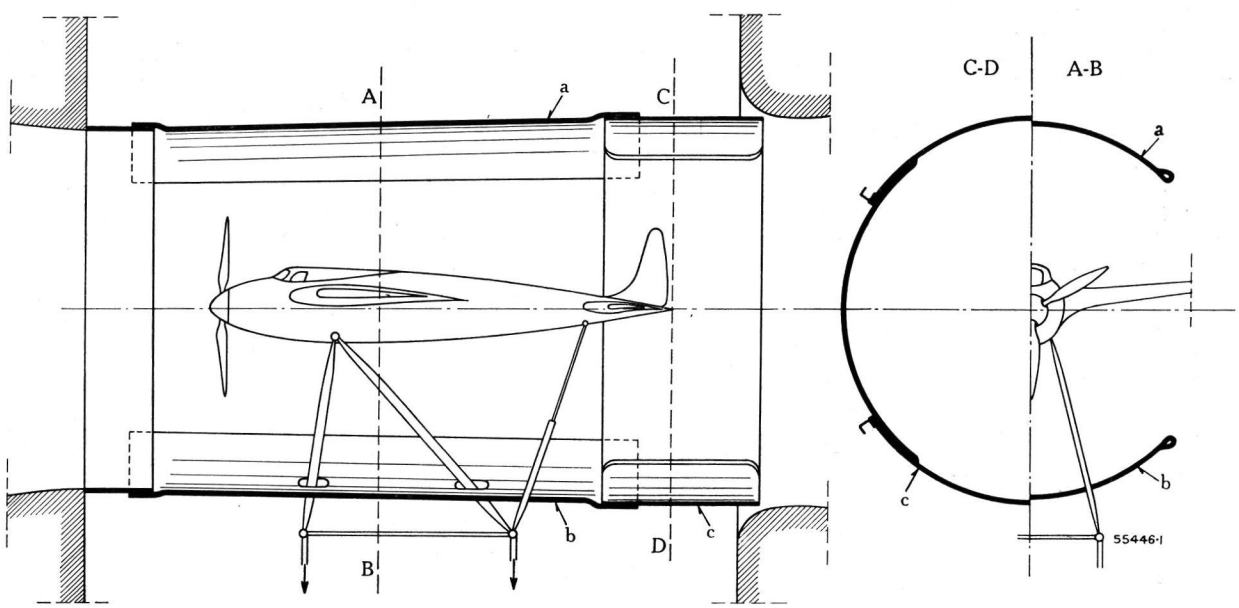


Fig. 9. — Wind-tunnel with partly guided jet.

This type of wind tunnel, whilst affording the same accessibility as the open jet type of tunnel, has a much smaller pressure drop, enables a better adjustment of the uniformity of the axial velocity and involves smaller corrections for taking into account the finite dimensions of the jet.

a. Removable ceiling. b. Removable floor. c. Adjustable sectors.

a simple and rigid design, consists of a set of radial partitions of stream-lined profile braced together by short stays in polygon form, so as to constitute a nest of pyramidal cells of approximately constant proportion, but increasing in size from the centre to the periphery.

Another simple design (Fig. 3) consists of a kind of cylindrical lantern with rectilinear and parallel vanes of constant width, disposed between the end wall of the building and the bell-mouthed inlet of the nozzle. Such filters ensure a good rectification of the air filaments and a low residual turbulence at the cost of a negligible pressure drop.

IV. THE PROPORTIONS OF THE JET.

The achievement of a uniform velocity in the jet where it passes through the test chamber, is due to the irrotational character of the motion at the outlet from the nozzle, bound up with the maintenance of a uniform static pressure in this region. For this reason tunnels with guided jets require for the compensation of the pressure drop, only a slight enlargement of the section. This enlargement may also be looked upon as destined to provide a passage about the uniform parallel central core for the thin but growing boundary layer, progressively built up by the retarding action of the friction at the walls. In the case of open jets, the conditions are more complex, because of the encroachment into the contour of the jet of the turbulent transition envelope allowing the initial irrotational undisturbed flow to exist only at the centre of a cone whose apex lies on the axis at a distance of about 4·5 times the diameter of the nozzle.

The maintenance of a uniform flow, at least in this inner zone of the jet, depends essentially on the conditions of enlargement of the jet between the nozzle and the diffusor. This latter section, which certain empirical relations have tried to express as a function of the length of the jet, is generally made too great, so that in the majority of existing installations there is an appreciable slowing-down of the velocity and a rise in pressure along the jet leading to an underestimation of resistance, notably of stream-lined bodies, or necessitating doubtful corrections.

Experience shows, and this is borne out by the application to this case of the semi-empirical theory* of the diffusion of Prandtl-Tollmien, that the correct enlargement for a free jet length equal to the diameter is 3·5 per cent.

It is, however, true that a larger section at the inlet to the diffusor, by allowing a reduction in length of this latter and of the kinetic energy to be recuperated therein, results in a higher efficiency of the installation and that regard for the size of the wake of the models, when the couple which they produce is appreciable compared with the size of the jet, also leads to a wider section, so that the most favourable enlargement should be finally chosen somewhat greater than the theoretical one (for example, of the order of 10 per cent for L=D).

The use of a variable inlet opening enables, moreover, a subsequent adjustment to be made. Such a solution which has been employed by Prof. Ackeret for the octagonal section of his wind tunnel at the Swiss Federal Institute of Technology, Zurich, can be applied to a circular section split up into four sectors (Fig. 9) and can be conveniently combined, both with the arrangement suggested above of suspended roof and ceiling for partially guiding the jet, and with the circular slot which sometimes has to be provided near the entry of the diffusor for preventing vibrations.

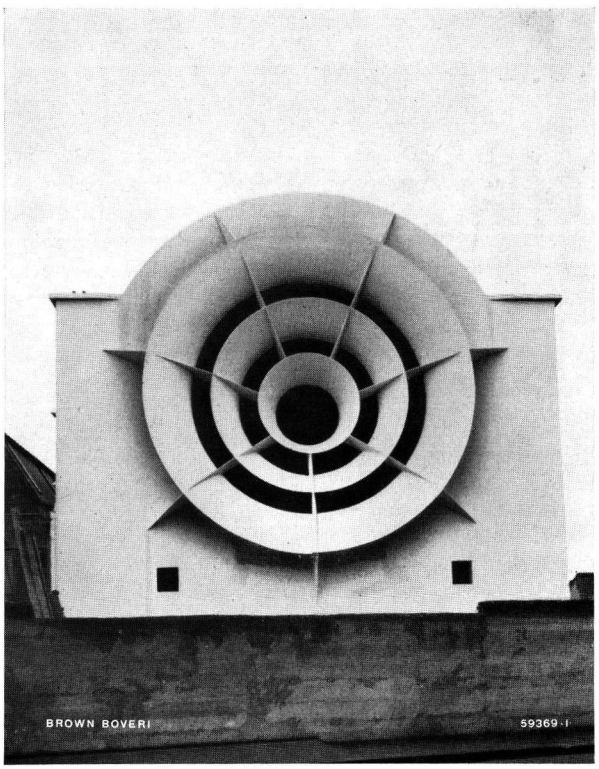


Fig. 10. — Diffusor outlet of a large wind-tunnel.
The circular deflecting vanes spread the air jet out and enable 80 % of the kinetic energy to be recuperated.

V. VIBRATION OF THE JET.

A phenomena often encountered at certain critical velocities in wind tunnels with open jets is a spontaneous vibration of the air column, which may occasionally attain a high intensity, and which, always troublesome, has to be overcome by some artifice or other. Although the mechanism of this phenomena has not yet been entirely elucidated, so that it is not possible to ensure its absence at the time of designing the tunnel, it is evidently related to the instability of surfaces of velocity discontinuity, already studied by Helmholtz, and appears to depend neither on the general design of the wind tunnel, nor on its natural resonance frequency, but practically exclusively on the

* A. M. Kuethe "Investigation of the turbulent mixing regions formed by jets". Journal of Applied Mechanics. Sept., 1935, p. A 86.

length of the test chamber, because the vibration frequency corresponds as a rule to the semi-wave length of this chamber (for example, 12 cycles, or 340 m/s: 3×14 for the chamber 14 m long of Fig. 1).

This vibration is either maintained or damped by a mechanism akin to that of organ pipes, depending on the phase at which the periodic alternating expansions and contractions originating at the outlet of the nozzle under the influence of the pressure variations along the contour of the jet occur at the inlet of the diffusor. These successive bunchings which in damp weather are well shown up at the periphery of the jet by a fog of condensed moisture, are propagated with a velocity approximately equal to half the jet velocity.

If no modification of the length of the chamber is possible, the necessary damping effect is commonly obtained by placing the chamber in communication with the diffusor by means of a circular slot in the latter.

As, however, the usual proportions of the diffusor are such that there is an over-pressure at this slot*, the resulting leakage represents a loss which tends to reduce the efficiency, but which on the other hand provides a means of controlling the uniformity of the jet. This loss, which is in no way connected with the effectiveness of the damping, can be reduced or even eliminated altogether, if, as already mentioned, the relative opening of the diffusor is diminished.

VI. PROPORTIONS OF THE DIFFUSOR.

It seems to be advantageous in the case of tunnels with an open jet, especially when the open portion is fairly long, to place in front of the diffusor, a short cylindrical section in which the profile of the velocities has the opportunity of adapting itself to the new conditions, and where a certain amount of the pressure is recovered. The opening of the diffusor, which is often fixed at 7^0, should in reality be made progressively smaller as the Reynold's number increases. As a compromise between the frictional losses, and those due to the inequality of the velocity profile, it may be as low as 5 or 6^0 in large wind tunnels and cannot be increased beyond 9^0 to 10^0 without entailing a serious sacrifice in efficiency**. Another objectionable consequence of an exaggerated opening is the accentuation of the inequality of the profile of velocities, which results in poorer operating conditions for the fan.

The very high efficiency of large diffusors, which for Reynolds numbers of the order of 10^7 attains 95 per cent, explains why the coefficient of use of high velocity wind tunnels (250 m/s), which can be achieved only with guided jets, may exceed 12.

The fact that these losses, although small, are still two to three times greater than the corresponding losses in a cylindrical tube, whereas the smaller velocity gradient along the normal to the walls would suggest an even lower value, is explained by the occurrence in the diverging duct of a turbulence anisotropy, which contrary to that taking place in a converging duct, magnifies the longitudinal component of the inequalities of velocity at the expense of the transversal ones, and, reducing the thrusts on the lateral walls and hence their components parallel to the axis, contributes to the pressure drop a force which is additive to the tangential frictional forces.

VII. THE FAN.

The preoccupation of reducing the diameter of the fan wheel and of keeping down the costs of the installation generally leads to placing the fan fairly close to the inlet of the diffusor, whereas in most cases the efficiency would be improved by moving the fan right back to the outlet because not only does the air screw contribute to the uniformity of the flow through the corresponding section, but the space taken up by the cowling about the motor has here a less unfavourable influence on the expanding effect of the diffusor than further upstream. Moreover, the ventilation system of the motor can, as shown in Fig. 1, also be employed to improve the conditions, by aspirating the boundary layer in the rear portion of the cowelling. The design of the air screw proper can be done very accurately according to the classical aerofoil theory, by taking into account the very unequal radial repartition of the total pressure drop, small at the axis, especially in the absence of a model, and large at the walls. The efficiency, crediting the fan with the mean radial value of the total pressure which it generates, may thus exceed 93 per cent.

VIII. THE DIFFUSOR OUTLET.

The energy corresponding to the residual velocity at the outlet of open-type tunnels, represents as a rule, an appreciable loss which may be reduced by a cumbersome extension of the slightly expanding section of the diffusor. A trumpet-shaped end of this latter does not bring any improvement in this respect, as the flow separates immediately from the wall; a better arrangement consists of an open tulip-shaped core, as an extension of the motor cowelling when the latter is placed at the diffusor outlet, but the best results are obtained by the use of a system of circular vanes of toroidal form, which by forcing the fluid to discharge out radially over a solid angle of nearly a hemisphere, ensures the recovery of approximately 80 per cent of the kinetic energy of the end section immediately preceding these vanes. In the case of the arrangement illustrated in Fig. 1, this device, in spite of its small space requirements, saves about 500 kW, or about 15 per cent of the power of the fan, in addition to diminishing considerably the inconvenience of the wind discharged from the tunnel (Fig. 10).

* This over-pressure is present, even if the diffusor section is constant up to the slot and the static pressure in the free part of the jet is uniform, due to the smoothing effect on the profile of velocities immediately on entering the diffusor, of the sudden reduction in friction at the contour of the jet, so that the leakage from the slot cannot be stopped except by making the first ring of the diffusor not merely non-diverging but slightly converging.

** "F. L. Wattendorf—Factors influencing the energy ratio of return-flow wind tunnels" 5th International Congress for Applied Mechanics, Cambridge, U. S. A. (1938), p. 536.

IX. CONCLUSION.

The history of wind tunnels is relatively short, but the role which they have played in the development of aeronautics has already proved to be a capital one, and the number and scale of installations planned and under construction (up to 40,000 kW) affords evidence that the importance attached to them is not likely to diminish in the future. While many improvements still have to be made, some of which such as the aspiration of the boundary layer which would enable the length of the diffusor to be reduced, can just be vaguely anticipated, others, such as the reduction of the turbulence, or the raising of the efficiency simply await general application, the present state of the art does, nevertheless, comprise a wealth of data and experience, often obtained at much cost or in the most unexpected fields, the correct appreciation and the safe application of which demands even more skill and science on the part of the designer than is required for the carrying out of tests.

The capital and interests involved in the creation of a large and powerful wind tunnel installation are so great that it is inadmissible to base on preconceived ideas or on conclusions hastily deduced from a few special tests of limited scope, and it is better to rely on the sound and large experience gained in a wide variety of fields, which only firms such as Brown Boveri possess.

(MS 987) *G. Darrieus. (Hv.)*

SUPERSONIC WIND TUNNELS.

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The flow of air at speeds above sound is essentially different from that below sound and requires special supersonic wind tunnels for investigation. Whereas the older installations could be operated intermittently only, there are to-day plants which can be operated continuously, and which are bringing a valuable contribution to the science of ballistics, aeronautics and turbo-machine design.

I. FLOW IN COMPRESSIBLE MEDIA.

THE path of a stream particle is determined by the forces of gravity, by inertia and viscosity. In the case of the high velocities common in aerodynamics, the effect of gravity is negligible and the motions depend mainly on the inertia and frictional forces. The stream lines of two different media, for instance, air and water, in geometrically similar channels or machines are similar at all points when the ratio of the inertia to the frictional forces in both systems is equal. This means that a definite ratio must exist between the dimensions d, the velocities w, the densities ρ and the viscosities η, namely equality of the Reynold's numbers:

$$Re = \frac{w\,d\,\rho}{\eta}$$

When this condition is fulfilled, the ratio of the frictional loss to the flow of energy and hence the efficiencies of the two systems are equal. It is, therefore, appropriate to express the efficiencies and loss coefficients as a function of the Reynold's number. But the example water—air shows that a limitation applies. Water is practically incompressible, air, on the other hand, varies in volume considerably, if the pressure changes appreciably. Appreciable changes of pressure and density always accompany high velocities.

The pressure and density conditions are conveniently characterized by the ratio $M = w/a$ of the velocity of flow to the velocity of sound, i. e., by *Mach number*[1].

It is no hazard that the velocity with which a small change of pressure or disturbance is propagated, appears in the flow equations of compressible media, since the propagation of a pressure or velocity wave depends on the compressibility.

The Mach's number has a further significance. It allows the flows to be divided into two categories. The subsonic and the supersonic regions, in which fundamentally different phenomena take place.

II. SUBSONIC AND SUPERSONIC FLOWS.

In order to understand the characteristics of flow phenomena below and above sound, we make use of the following concept:

A fluid in motion is guided by the wall enclosing it. We may imagine this guidance to be in the form of a large number of small impulses or blows coming from the elements of the wall. This picture is by no means entirely arbitrary if the molecular structure of matter is recalled. Every small blow produces in the stationary gas a spherical wave, which travels away from

[1] If at any point P_0, where the flow is stationary, the density is ϱ_0, and at some point P, there is a velocity w and a density ϱ, then there exists between the velocities and the densities the following relation:

$$\left(\frac{w}{a}\right)^2 = \frac{2}{k-1}\left[\left(\frac{p_0}{p}\right)^{\frac{k-1}{k}} - 1\right] = \frac{2}{k-1}\left[\left(\frac{\varrho_0}{\varrho}\right)^{k-1} - 1\right]$$

where a is the velocity of sound at point P.

This expression follows from the energy law:

$$\frac{w^2}{2g} = c_p\,(T_0 - T)$$

and the expression for the velocity of sound:

$$a^2 = g \cdot k \cdot R \cdot T$$

and the adiabatic equation of state:

$$\frac{T}{T_0} = \left(\frac{p}{p_0}\right)^{\frac{k-1}{k}} = \left(\frac{\varrho}{\varrho_0}\right)^{k-1}$$

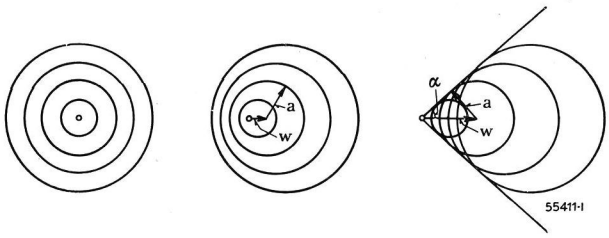


Fig. 1. — Propagation of a disturbance wave in a stream.

Rest	Below sound	Above sound
$w = 0$	$w < a$	$w > a$
		$\sin \alpha = \frac{a}{w}$

The fundamental difference between subsonic and supersonic speed is clearly visible.

the centre of disturbance at the velocity of sound, the amplitude diminishing naturally with the distance.

If the gas is in motion relatively to the wall element, the wave is carried along with the stream. It is no longer centred around the focus of the disturbance (Fig. 1). If the gas flows with a velocity above that of sound, the wave is no longer able to travel up against the current. It is swept along in one direction and appears only within a cone whose generatrix makes with the direction of flow the Mach angle: $\left(\sin \alpha = \frac{a}{w} = \frac{1}{M}\right)$.

Outside this cone it is not possible to detect any influence on the flow. Thus entirely different forms of flow and quite different flow conditions exist in the zones below and above the velocity of sound.

Fig. 2 shows for instance the flow about a strut. At velocities below sound there is an increase in speed in the neighbourhood of the thickest part. The reduction in pressure allows the specific volume of the gas at this point to increase. This, in turn, produces a further displacement of the tubes of flow. The disturbance caused by the strut is, therefore, transmitted farther into the stream. The ratio of the forces acting on the strut to $\frac{\rho}{2} w^2$ increases with increasing velocity, the theoretical increase being in the ratio of $1 : \sqrt{1-M^2}$. Similarly the lift of an aerofoil, making a small angle of incidence with the direction of flow, increases in the same ratio. In actual fact experiments show that this increase in force goes on up to a certain Mach number. The curve then breaks away suddenly because various secondary effects begin to become manifest (Fig. 3).

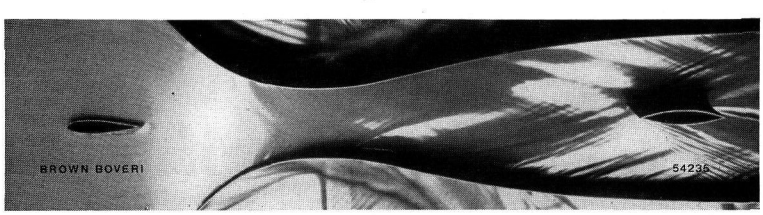


Fig. 2. — Flow around a strut

below and above sound velocity illustrated in an open water channel.

At velocities above sound every change in direction of the wall, gives rise to a wave making the Mach angle with the direction of flow. A first pair of waves arises from the nose of the strut. The stream lines are refracted at the wave front. By applying the laws of refraction it is possible to draw a supersonic field of stream lines agreeing closely with that obtained by experiment[1].

Special conditions exist at the boundaries of the two regions, i. e., just where the limit of sound is crossed. The mere fact that the above-mentioned force ratio $1 : \sqrt{1-M^2}$ becomes infinite when $M = 1$ points to an anomaly.

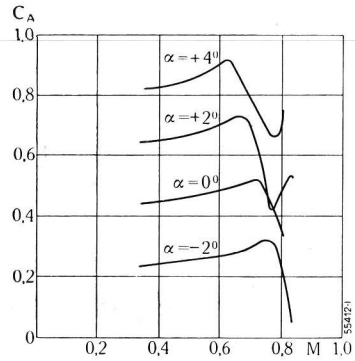


Fig. 3. — Lift of an aerofoil as a function of the Mach number (according to Durand).

It can be seen how the lift coefficient C_A falls rapidly upon approaching the velocity of sound.

We shall now pursue our picture of the effect on the flow of small disturbances a stage further.

Fig. 4 shows a field of flow with the velocity decreasing from left to right. The velocity of sound is crossed at the line A. Let O be the source of small disturbing waves, which travel out in all directions, i. e., also against the direction of flow. On arriving at the boundary A, they are unable to penetrate further against a flow velocity greater than sound. The result is an accumulation of these disturbances on the line A, which combine to form a discontinuity in the pressure and in the velocity. The transition from supersonic to subsonic occurs almost always discontinuously with a compression shock which is one of the typical characteristics of supersonic flow. The laws of these compression shocks are well known. We shall make use of them in our treatment of the wind tunnel.

It is easily seen that for the converse case of flow from the subsonic to the supersonic region, there is no cause for

[1] The method is due to Prandtl and Busemann, see Stodola Festschrift 1929, p. 499 or Prandtl: Abriss der Strömungslehre, p. 203.

an accumulation of disturbances. The waves coming from the subsonic region are simply swept over the boundary, and no waves from the supersound region can approach the boundary.

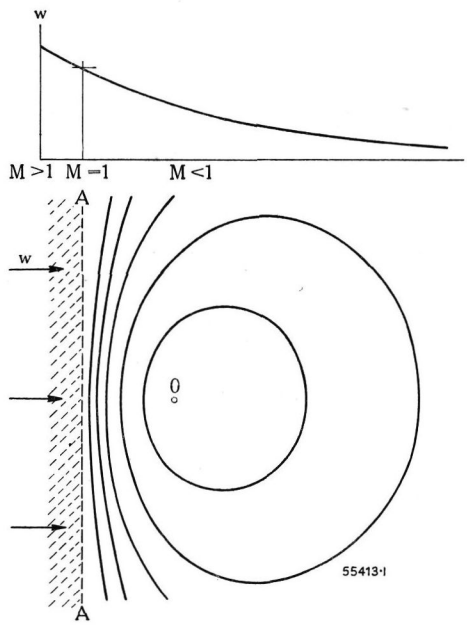


Fig. 4. — Transition from infra- to super-sonic velocity.

Creation of the compression shock due to the fact that the disturbance waves, which originate for instance at O, cannot penetrate upstream beyond the limit A—A and therefore accumulate at this point.

Another surprising difference between supersonic and subsonic flow is to be found in the flow of fluid with friction through a cylindrical tube.

With increasing Mach number the ratio of the pressure drop to the kinetic energy increases rapidly. It becomes infinite at sound velocity, from which it follows that it is impossible for a fluid to flow through a cylindrical tube at the velocity of sound. At velocities above sound the pressure drop turns into a pressure rise. The energy loss, which of course occurs just as in the previous case, is entirely at the expense of the kinetic energy[1].

III. SUPERSONIC WIND TUNNELS.

A few examples have shown us the essentials of supersonic flow. It is, therefore, clear that the experience obtained in subsonic wind tunnels does not allow any deductions to be made as to the behaviour of aerofoils, turbine blades or projectiles in the domain above sound.

Basically, the carrying out of a model test at high velocity requires that both the Reynold's and the Mach numbers shall be the same for the model as for the full scale apparatus.

The power of the supersonic jet is quite considerable and amounts to

$$L_w = F \cdot \frac{\varrho w^3}{2} = \lambda\,(M)\, F \frac{\varrho_0 a_0{}^3}{2}$$

where F is the area of the test section, ϱ the density in the test section and w the velocity in the test section; ϱ_0 and a_0 denote the density and the sound velocity at the state in front of the nozzle and is a function of the Mach number, which can be taken from the diagram Fig. 5. The adiabatic compression work L_k, to be supplied by a compressor, in order

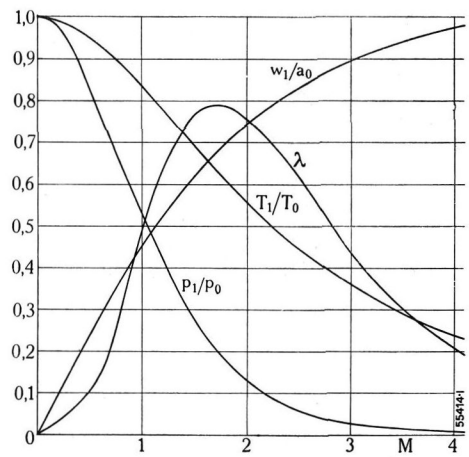


Fig. 5. — Diagram of expansion nozzle (no loss).

p_0, T_0, a_0. Pressure, temperature and sound velocity in front of the nozzle.
p_1, T_1, w_1. Pressure, temperature and velocity of flow after the nozzle as a function of the Mach number M.
λ. Power coefficient: $\varrho_1 w_1{}^3 = \lambda\, \varrho_0 a_0{}^3$.

to recompress the air from the final pressure at the nozzle, after entire loss of the kinetic energy, is greater than L_w in the ratio T_0/T (Fig. 5). For instance, for an initial condition $P_0 = 1{\cdot}0$ kg/cm² abs, $T_0 = 300^0$ K, the powers L_w and L_k per m² of test section are:

M	1	2	3
L_w	12,000	18,000	10,400 kW
L_k	14,400	32,500	29,300 kW
R_e	15	9·6	$4{\cdot}0 \times 10^6$

The Reynold's numbers are referred to a square test section of 1 m length of side, whereby it should be remembered that the Reynold's numbers referred to some dimension of a model located in the test section will be correspondingly smaller. The power actually required is a fraction 1/q of L_k and depends

[1] For incompressible media we have the relation:

$$\Delta p = -\lambda \frac{l}{d} \frac{\varrho w^2}{2}$$

For compressible media this must be replaced by:

$$\Delta p = -\frac{1+(k-1)M^2}{1-M^2} \lambda \frac{l}{d} \frac{\varrho w^2}{2}$$

an expression which becomes infinite for $M = 1$ and which changes its sign when M becomes greater than 1.

on the amount of the kinetic energy which can be recuperated after the test section, and does not attain such small values in supersonic tunnels as in subsonic tunnels.

Two methods of reducing the energy requirements have been applied:—

1. By the quick discharge of an energy accumulator.
2. By reducing the pressure in continuously operating plants.

The older channels, for instance the first Göttingen installation, is of the accumulator type. A receiver of 10 m^3 content is evacuated and hereafter filled via the test section of 6×6 cm to atmospheric pressure, during which a supersonic velocity can be maintained

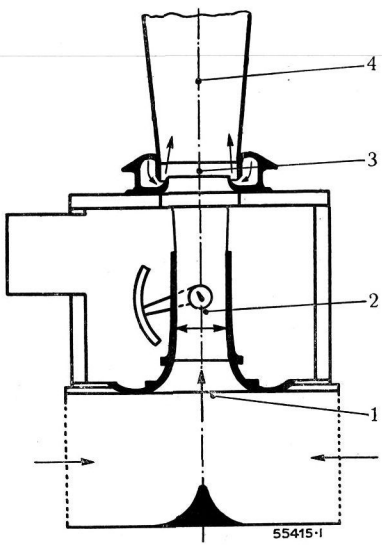


Fig. 6. — Supersonic wind tunnel with compressed air accumulator (U. S. A.).

1. Nozzle. 3. Ejector supplied by accumulator.
2. Measuring section. 4. Diffusor.

Such tunnels can only operate intermittently.

for 10 seconds. Other channels work with a pressure receiver which, in order to produce the desired vacuum, discharges into an ejector beyond the test section. To this type belongs a tunnel with a test section of 30 cm diameter in England and one of 60 cm diameter in America (Fig. 6). In a Russian installation the pressure receiver discharges through a reducing valve directly into the test nozzle. The temperature in the test section is kept under control by the injection of fuel before the nozzle. This is the only plant where the pressure and temperature in the test section reach atmospheric conditions at sea level. Clearly the new types of supersonic wind tunnels with continuously working plants are much to be preferred. The closed type consisting of a compressor, cooler, test chamber and return channel, has — compared with the open suction type — the great advantage that it is possible to regulate the level of the absolute pressure in the system by means of a pump, and hence to adjust the power requirements to the available source. A higher density with a smaller test section leads with a given power to higher Reynold's numbers than a small density with a large test section. However, it is not desirable to reduce the size of the test section below a certain minimum, especially when it is considered that in the neighbourhood of the velocity of sound the model should be small in proportion to the size of the tunnel. Moreover, the suspension of the model becomes more difficult with increasing density.

A particular advantage of the closed tunnel consists in the fact that by adjusting the pressure, the Reynold's numbers can be varied, while maintaining the same Mach number. Generally, it will never be possible to reach the large Reynold's numbers encountered in practice. All the same it is important to test whether within the attainable limits, as is generally assumed, the effect of the Reynold's number at high Mach numbers is of secondary importance. Further, the closed tunnel allows the use of gases other than air, such as monatomic or heavy gases, for which the laws of flow are different from those for air.

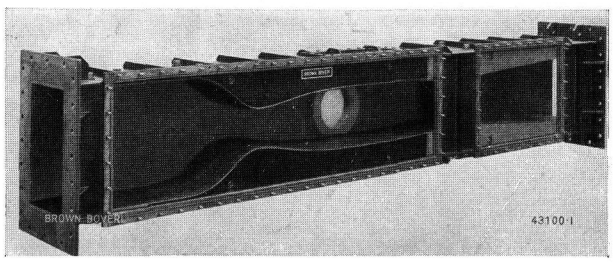


Fig. 7. — Test section with cover removed.

The section comprises a convergent-divergent nozzle, a parallel walled measuring chamber of rectangular section and a convergent-divergent diffusor.

IV. THE TEST SECTION.

In the test section the air current has to be accelerated to the required velocity in the test chamber. Beyond the test chamber the kinetic energy should be converted back into pressure at as high an efficiency as possible. Fig. 7 shows such a test section with the following main parts:—

(a) Converging nozzle for accelerating the flow to sound velocity.

(b) Diverging nozzle for further acceleration of the flow above the velocity of sound.

(c) Measuring chamber in which the lines of flow must be as nearly as possible parallel.

(d) Converging part of the diffusor for reducing the velocity down to that of sound, or more exactly, as we shall see, down to a value still in the supersound region.

(e) Diverging part of the diffusor for further reducing the velocity down to a value suitable for the pipe systems, and which is far below that of sound.

In principle, it should be possible behind the test section to reverse the sequence of conditions in the nozzle in a converging-diverging diffusor symmetrical to the nozzle. Experience shows, however, that the second neck must be greater than the first one by a definite amount, even with perfectly frictionless flow.

Let us imagine that the tunnel is put into operation in such a way that the pressure at the entrance of the test section is kept constant, whilst it is progressively lowered at the outlet. If now the second neck

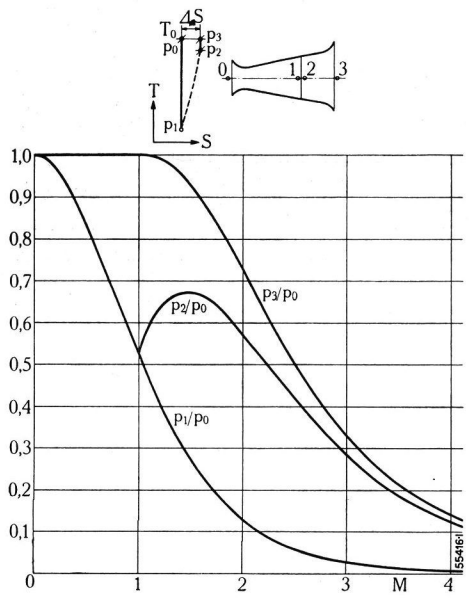


Fig. 8. — Laws of the compression shock.

p_0. Pressure in front of the nozzle.
p_1. Pressure in front of the shock wave.
p_2. Pressure behind the shock wave.
p_3. Pressure after recuperation of the kinetic energy beyond the shock wave.

is narrower than the first one, the sound velocity appears there first. From that moment onward, the entire flow upstream cannot be effected by a further lowering of the pressure and it is impossible to obtain a supersonic velocity in the test section (Fig. 9a).

If on the other hand, the first neck is narrower, then the velocity of sound occurs first there (Fig. 9b). Thereupon a zone of supersound velocity is set up in the diverging nozzle, which changes to subsonic in a compression shock. With decreasing final pressure, the shock wave is displaced up towards the test section and is suddenly swept down to the diverging part of the diffusor, where it becomes stationary (Fig. 9c). Further lowering of the final pressure has no effect on the test section. It is even possible to increase the back pressure slightly, the shock wave moving up towards the second neck, which, however, it must not be allowed to reach, as it would then spring over into the nozzle.

The sudden compression at the transition point involves an increase in entropy. The state of the gas after this shock is represented by another isentropic line which may be determined from the fact that if the original temperature T_0 is restored by a frictionless recuperation of the velocity, the pressure p_3 will be lower than the initial value p_0 (Fig. 8).

This has important consequences. If after a compression shock the jet is once again accelerated, the sound velocity takes place in a section different from that before the shock, where the area is p_0/p_3 times as great. If now the second neck in our test section is not larger by at least this amount than the first one, a second sound velocity zone occurs there, whilst the first compression front still lies in the diverging part of the nozzle. It is not possible to get this compression front to pass over the point where it is swept down the test section (Fig. 9d).

The second neck must, therefore, be greater than the first one, at least in the ratio p_0/p_3, to which a margin should be added for the increasing thickness of the boundary layer and to insure stable operation.

Stable operation of the test section is, therefore, characterized as follows:—

Production of the supersound velocity in the convergent-divergent nozzle, reduction of the velocity in the converging diffusor to a value slightly above that of sound, a small re-increase in the diverging diffusor, a sudden jump to subsonic velocity in the shock front and further reduction in the rear part of the diffusor. It is not surprising that the degree of recuperation of the kinetic energy decreases with increasing Mach number. At the moment when, during putting into service, the shock front reaches the test section, the second neck offers no further advantages as only subsonic speed exists therein. The compressor must give a pressure ratio just as if no converging-diverging diffusor were present, even if the load can be subsequently reduced. In spite of this a purely diverging diffusor would not be advantageous, because in order to keep the shock front away from the test section, it would be necessary to go to a velocity still higher than that in the test section, in part of this diffusor which would result in larger shock losses.

The difficulty of the pressure jump at the moment of starting up can be avoided if the supersound speed is built up in a nozzle with adjustable elastic walls instead of in one with fixed walls, such as has been recently done by Professor Ackeret. Not only does such a nozzle allow the Mach number to be varied continuously during the tests, but it also allows the ratio of the size of the second neck to the first one to be reduced and ensures a better recuperation of energy.

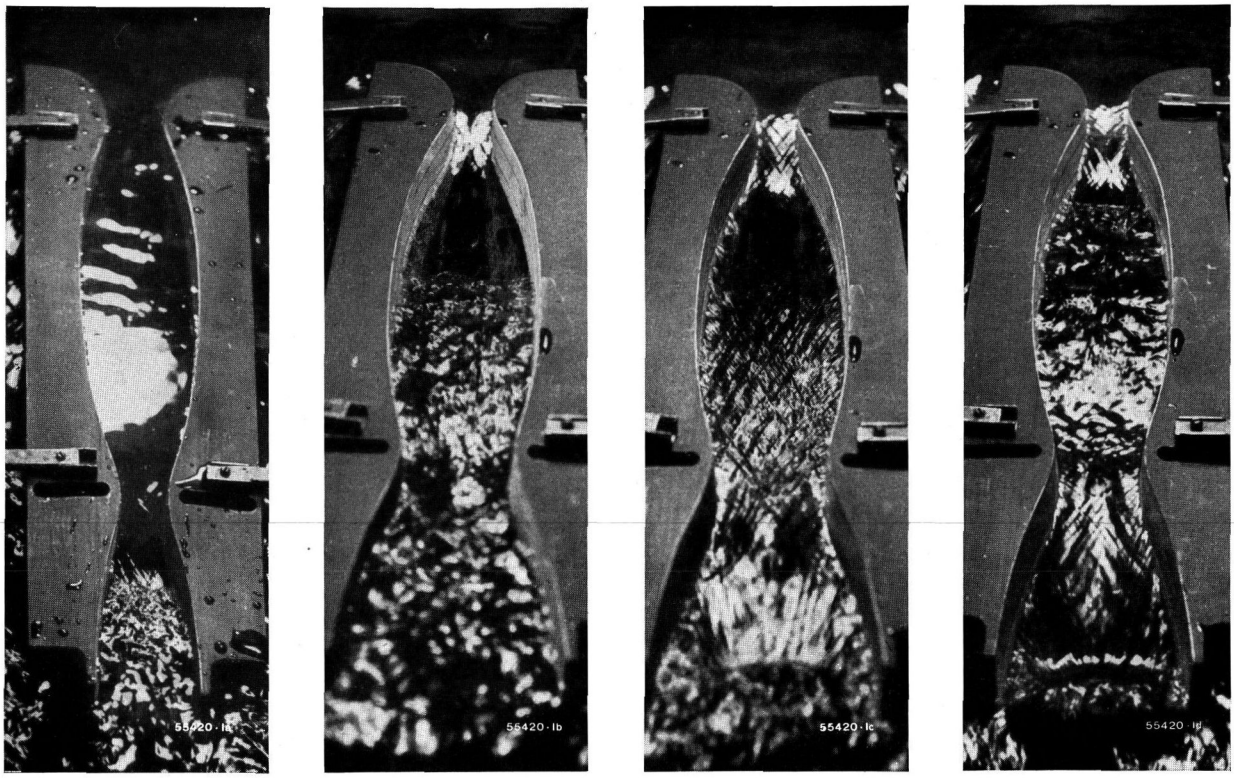


Fig. 9a – d. — Representation of the phenomena in the supersonic test by means of the flow of water in an open channel.

a. Appearance of the supersonic velocity behind the second, narrow neck.

b. Appearance of the supersonic velocity behind the first neck during putting into service of a correctly dimensioned measuring section.

c. Supersonic velocity in the measuring section. Compression shock behind the second neck (normal operation).

d. Appearance of two pressure shocks, one after the first neck, the other after the second neck. Velocity in the test space below that of sound. The second neck is larger, but only slightly larger than the first one.

In order to achieve as nearly as possible parallel flow in the test section, the nozzle must be most carefully designed. The shape of the diverging part, in which the flow takes place above the velocity of sound, must in particular be given most careful attention. Every change of direction of the wall causes Mach waves to be set up. The waves, which originate in the neighbourhood of the neck of the nozzle travel across the channel and reach the opposite wall. It is necessary to ensure that at the point where they reach the opposite wall, a change of direction takes place, which sets up a wave of opposite sign and which, compounded with the first one, cancels it out. The waves originate at the convex wall and are destroyed at the concave one. The outline of a rectangular nozzle contained between two parallel walls can be theoretically determined by the Prandtl-Busemann method (Fig. 10). An analogous method has been developed by Darrieus for circular nozzles.

The equipment of the test section includes a precision barometer, measuring low absolute pressures. The suspension arrangement between the model and the balance is heavily loaded and must be located at a suitable place, for example, along the extension of the axes of a projectile model, in order that the waves shall not be disturbed. The suspension wires must be surrounded by fixed shrouds to protect them from wind pressures.

The conditions in the field of flow can be made visible by means of the Töpler method (Fig. 11). The field is illuminated, the light rays being refracted by the variations of pressure and throw a picture on the screen of the density gradients in the flow. In this manner it is possible to make visible Mach waves, shocks, separations, eddies, etc.

V. THE BLOWER

The pressure ratio to be delivered by the blower depends on the Mach number desired in the test section, the recuperation in the diffusor and the total resistance of the circuit. The following table is based on small scale nozzles, tested some by Prof. Ackeret and some by Brown Boveri, and gives conservative figures which may be obtained with certainty.

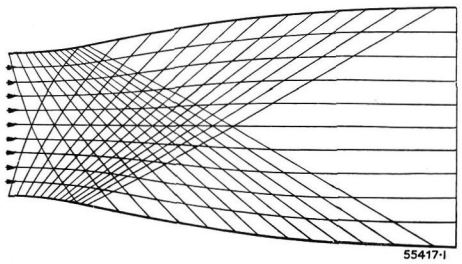


Fig. 10. — Supersonic nozzle.

Determination of the outline of the divergent part of the nozzle for producing an eddy-free parallel flow in the test chamber by the Prandtl-Busemann method.

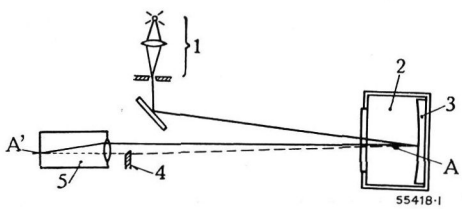


Fig. 11. — Toepler process
for making air flow visible.

1. Slit.
2. Measuring chamber in the wind tunnel.
3. Convex mirror.
4. Diaphragm.
5. Camera.

An upward rise of density at A deflects the light ray reflected by the convex mirror in an upward direction. The ray travels past the diaphragm 14, to the camera. The image A' of A appears light. A rise in density from bottom to top deflects the light ray in a downward direction. It is intersected by the diaphragm and A' appears dark.

Mach number	1	1·55	2	2·73
Pressure ratio of the nozzle .	1·89	3·96	7·82	24·2
Pressure ratio of the diffusor	1·205	1·39	1·78	3·15
Pressure ratio of the compressor	1·265	1·46	1·87	3·3
Adiabatic compression power of the blower: Kinetic energy of the jet	0·42	0·35	0·45	0·68
Adiabatic compression power of the blower power without recuperation	0·35	0·24	0·25	0·27
Volume at suction of the blower for 1 m^2 test section, m^3/s .	251	237	220	202

Thanks to its compact construction the Brown Boveri axial blower (Fig. 12) is particularly well suited to wind tunnels. It avoids any unnecessary change of direction of the air and can be easily incorporated in the duct circuit. It is, therefore, not by chance that the construction of the large supersound wind tunnel at the Swiss Federal Institute of Technology, Zurich, provided the opportunity for realizing the first multi-stage axial blower[1]. Fig. 13 shows the characteristics of a thirteen-stage blower. The measuring sections and the Mach number, which can be obtained therewith, have been inserted in the P-V diagram, and give a good picture of the possibilities of application of the blower. With small measuring sections the pumping limit is approached.

[1] Cf. The Brown Boveri Review 1936, p. 57.

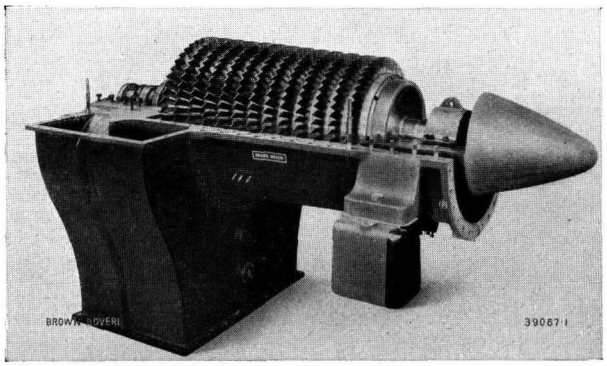


Fig. 12. — Thirteen-stage axial blower for the Zurich wind tunnel.

The first multi-stage axial blower built by Brown Boveri at the conclusion of a long period of development. This type of construction has since proved its worth in numerous applications.

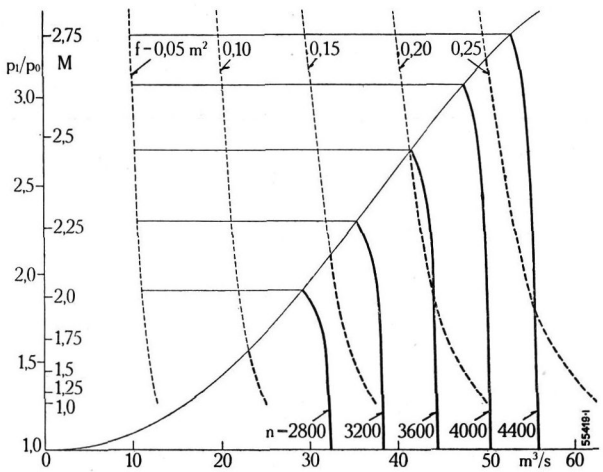


Fig. 13. — Characteristics of a multi-stage axial compressor.

The diagram shows the Mach number which can be obtained and the corresponding sections of the jet f.

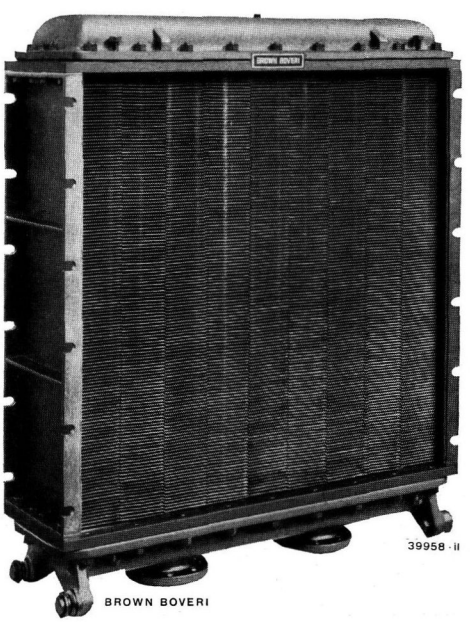


Fig. 14. — Cooler element.

The cooler fins serve at the same time to stabilize and to rectify the flow of the jet.

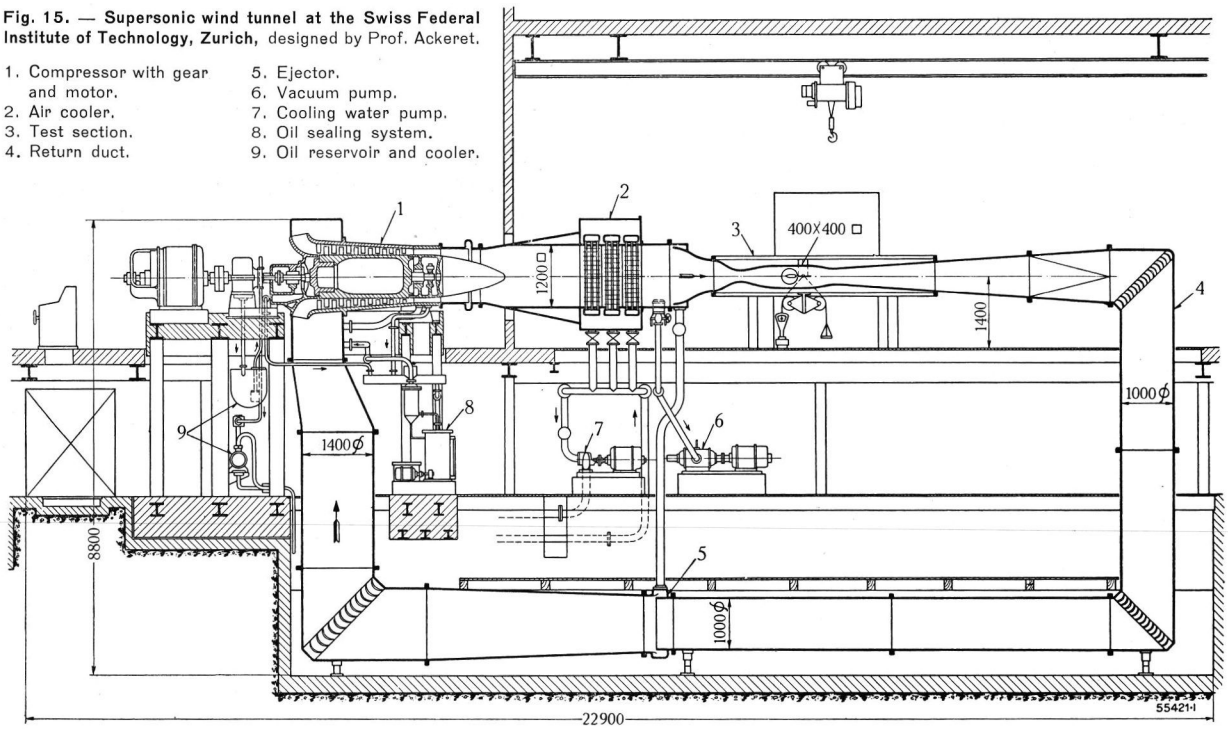


Fig. 15. — Supersonic wind tunnel at the Swiss Federal Institute of Technology, Zurich, designed by Prof. Ackeret.

This difficulty can be overcome by allowing part of the air to by-pass the measuring section by means of an adjustable valve. It is possible to use this by-pass air in an ejector to increase the pressure drop over the measuring section, and hence the Mach number. The gland seal of the blower shaft must be absolutely air-tight as the tunnel operates under vacuum. For this purpose oil of the bearing lubricating system is used as a sealing medium.

VI. THE COOLER.

In a high-power closed wind tunnel the heat radiation from the walls of the tubes is insufficient, and it is necessary to employ a cooler to keep the temperature down. It is convenient to use this cooler at the same time as a stabilizing honeycomb. It is inserted between the blower and the measuring section, where the temperature is greatest. Fig. 14 shows an element of the cooler. According to the power, either one cooler element, or several elements, one behind the other, are used.

In the case of high Mach numbers the air is also cooled between the compression stages.

VII. INSTALLATIONS BUILT BY BROWN BOVERI.

Figure 15 shows the installation at the Swiss Federal Institute of Technology, Zurich, designed by Prof. Ackeret, and which might be looked upon as the prototype of modern designs. Blower and motor are located in an engine room, cooler and measuring section in the laboratory proper. The cellular-type vacuum pump and the oil and cooling water pumps are in the basement. — A tunnel of similar dimensions was chosen by the Guidonia Institute near Rome. It has, however, a higher power available and can, therefore, operate with higher densities, thus presenting considerable advantages. A number of further supersonic wind tunnels are at present under construction. They will in due course find application in every large aerodynamic research institution as a necessary adjunct of the highly developed infra-sound tunnels.

(MS 926) *C. Seippel. (Hv.)*

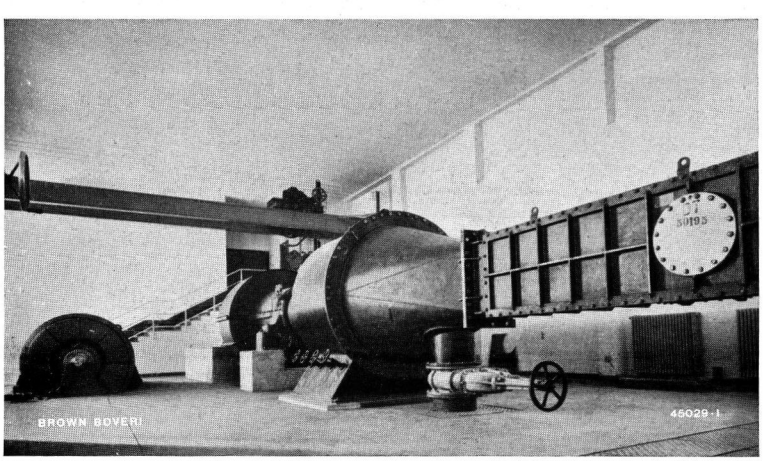


Fig. 16. — Supersonic wind tunnel at Guidonia near Rome.
The power is about three times as great as that of the Zurich plant.

THE ADAPTATION OF TURBO-CHARGED PRODUCER-GAS-OPERATED ENGINES TO VEHICLE RUNNING CONDITIONS.

Decimal Index 621.43.052.068.2 : 621.431.73

The evolution of the Brown Boveri turbo-charging process for producer-gas-operated engines is discussed and the simple method of obtaining a favourable torque characteristic for road vehicles described. The advantages of turbo-charging are also exemplified.

IN view of the present shortage of petrol and Diesel oil many road vehicles can only be kept in service by resorting to gas operation. The change-over to solid, indigenous fuels, however, frequently involves a big drop in power, which in the case of Diesel engines may attain 30 % and in the case of petrol engines 50 % or even more. As leading manufacturers of superchargers for Diesel engines we were naturally interested in the application of exhaust turbo-chargers to producer-gas-operated engines.

Whereas for the supercharging of Diesel engines only the air has to be compressed, gas operation involves compression of both gas and air. Nevertheless, in order not to have to provide more than one charger per engine we fit the charging unit ahead of the gas producer, which is thus also supercharged. The consequent higher rate of gas flow achieved permits the same size of producer to be retained for the higher engine output. The present standard layout of engine installations with Brown Boveri turbo-chargers is shown in Fig. 1.

In contradistinction to the Brown Boveri process other systems employ two separate chargers for gas and air or the gas-air mixture is compressed in a single supercharger. Inasmuch as these sets are fitted after the gas producer they quickly get dirty and are subject to rapid wear due to the combined action of dust and moisture.

Apart from the development of a special type of turbo-charger the behaviour of the gas producer at

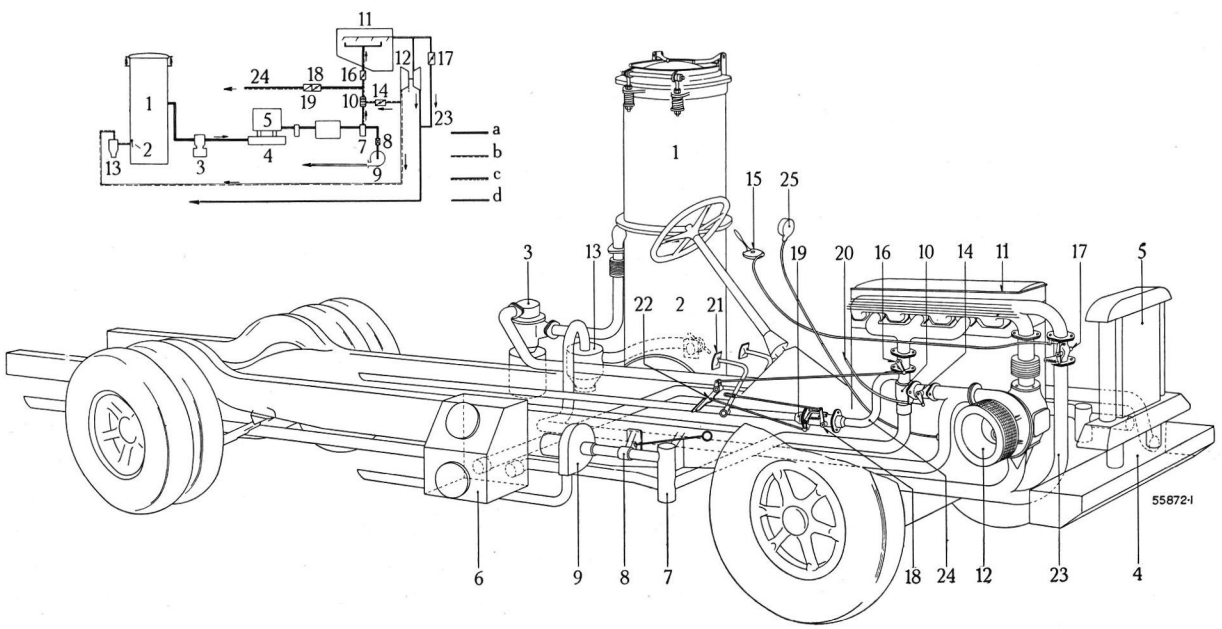


Fig. 1. — Diagram and perspective view of vehicle producer-gas installation with exhaust-gas turbo-charging.

It is only with an arrangement like the above that the original running properties can be retained.

1. Gas producer.
2. Non-return valve with ignition aperture cover.
3. Cyclone cleaner.
4. Baffle-plate precipitator-type cleaner.
5. Gas cooler.
6. Fine filter.
7. Condensed-water trap.
8. Stop valve.
9. Starting fan.
10. Mixing nozzle.
11. Internal combustion engine.
12. Exhaust turbo-charger.
13. Tar separator.
14. Air throttle valve.
15. By-pass valve control.
16. Gas throttle valve.
17. By-pass valve.
18. Mixture release valve.
19. Non-return valve.
20. Air throttle valve control.
21. Clutch pedal.
22. Mixture valve pedal.
23. By-pass pipe.
24. Mixture release pipe.
25. Pressure gauge.

a. Gas pipe. b. Air pipe. c. Mixture pipe. d. Exhaust pipe.

a gauge pressure of 0·3—0·4 kg/cm² also had to be investigated. Although the supercharged gas producer proved entirely satisfactory, certain difficulties, connected with the characteristics of the engine and charger, were encountered out on the road.

A traction engine should preferably have a similar torque-speed characteristic to a reciprocating steam engine, i. e., a hyperbolic characteristic, when delivering

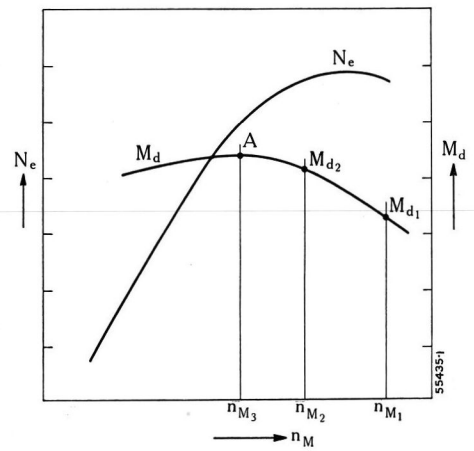


Fig. 2. — Power and torque curves of internal combustion engine in function of revolutions.

N_e. Engine output in B. H. P.
M_d. Engine torque in mkg.
n_M. Engine revolutions.

Even with supercharging the engine torque can be made to increase with falling revolutions over a wide range Δn_M by correctly proportioning the exhaust turbo-chargers.

an approximately constant output. This is an inherent characteristic of the steam engine through the fact that its energy is obtained in the form of steam from an extraneous source, whereas the internal combustion engine first has to prepare the fuel in a cylinder and convert it into heat through combustion. In consequence, the output and torque curves of an internal combustion engine (shown in Fig. 2) deviate considerably from the desired characteristic, and torque convertors are therefore indispensable for traction applications.

If a vehicle with an engine having the torque characteristic shown in Fig. 2 runs on to a gradient from the level at, for instance, the top speed V_1 corresponding to the engine revolutions n_{M1}, the speed will drop to V_2 or the revolutions to n_{M2}. Inasmuch as the engine revolutions are directly proportional to the speed of the vehicle they will drop from n_{M1} to n_{M2}, whereas the torque rises from Md_1 to Md_2. At n_{M2} revolutions the engine will thus automatically adjust itself to a definite new condition corresponding to the increased work involved by the gradient. With increasing tractive resistance the engine revolutions go back still further, e. g., as far as n_{M3}. Stable operation, however, is impossible on the drooping portion of the torque curve to the left of point A coinciding with diminishing revolutions. To prevent the engine stalling, therefore, the torque on the wheels must be stepped up in a convertor.

In the case of vehicles the convertor is usually in the form of a gear with three to five fixed ratios.

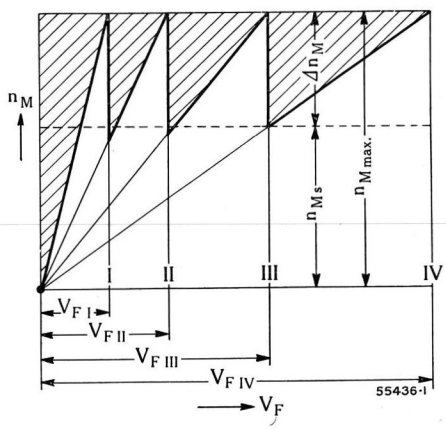


Fig. 3. — Gear-changing diagram.

n_M. R. P. M. of engine.
V_F. Speed of vehicle in km/h.
n_{Ms}. Gear-changing speed in r. p. m.
n_{Mmax}. Maximum r. p. m. of engine.
I. First gear. II. Second gear. III. Third gear. IV. Fourth gear.
V_{FI}, V_{FII}, V_{FIII}, V_{FIV}. Maximum running speed in first, second, third, or fourth gear.
The hatched triangles in the diagram represent the portions of the engine power range not utilized.

To achieve progressive variation of the driving wheel revolutions the intervals between the individual gear ratios must be bridged by altering the revolutions of the engine. The running speeds corresponding to the engine revolutions for the various ratios are well illustrated in Fig. 3. With one of the given ratios maximum engine revolutions correspond to a definite running speed which can be raised by changing to the next gear ratio. To ensure synchronization of the small pairs of gear-wheels, and due to the reduction of the ratio, the engine revolutions must be diminished, whereby, however, the engine output will also be decreased. The hatched triangles in Fig. 3 represent the portions of the engine power range not utilized. A gear giving an infinite number of ratios would be the most suitable because these triangles would then disappear, thus resulting in a torque with the desired characteristic. For practical reasons, however, only three to five ratios are employed. This has the double advantage of keeping the gear simple and not unnecessarily complicating matters for the driver. It is left to the engine, therefore, to adapt the torque to the

running conditions between the individual ratios. In consequence, it must be designed to have a rising torque with falling revolutions in the range of revolutions Δn_M between the individual gears. The engine can thus elastically cope with tractive resistance variations, and as long as they are not excessive, automatically smooth them out.

The same requirement must also be fulfilled in order to be able to accelerate a vehicle uphill. It is only possible to change gear upwards when after the gear change, at the inevitable jump in engine revolutions, the engine torque is the same as, or greater than, prior to the gear-changing operation.

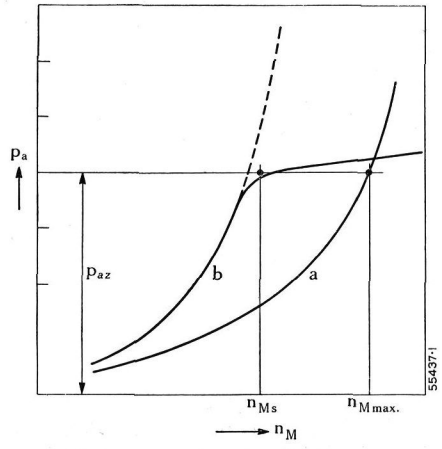


Fig. 4. — Charging pressures in function of the engine revolutions with and without regulation.

p_a. Charging pressure in kg/cm².
p_{az}. Admissible charging pressure in kg/cm².
a. Curve of charging pressure without regulation, in kg/cm².
b. Curve of charging pressure with regulation, in kg/cm².
n_{Ms}. Gear-changing revolutions.
n_{Mmax}. Maximum engine revolutions.
n_M. Engine revolutions.

A charging pressure characteristic as per curve *b* can only be achieved with an exhaust turbo-charger having a high overall efficiency.

round about the rated revolutions of the engine and with full charging pressure. A falling off of engine and charger revolutions consequently resulted in a heavy drop in output and torque, i. e., a torque characteristic drooping to the left.

The problem, therefore, was to find a way to correct the torque characteristic of supercharged engines to resemble as far as possible that obtained with liquid fuels. Instead of building the charger to attain the admissible charging pressure at maximum engine revolutions (curve a in Fig. 4) it is designed so that this pressure (curve b) is reached already at the gear-changing revolutions n_{MS}, shown in Fig. 3 by the dash

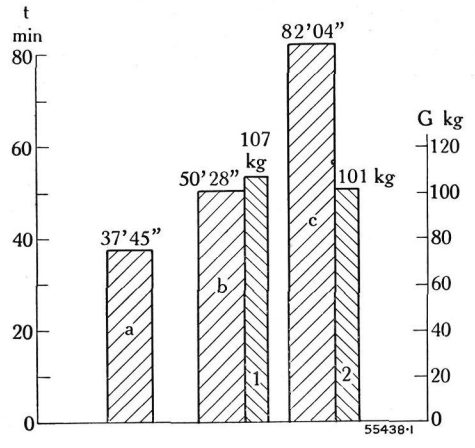


Fig. 5. — Running-time diagram of Opel-Blitz car with 3·6 litre engine over hilly route.

a. Running time on petrol.
b. Running time on wood-gas, turbo-charged.
c. Running time on wood-gas, non-supercharged.
1, 2. Wood consumption in kg/100 km.

The times apply to a 22 km run over a hilly route.

An interesting fact is that, due to the 38·5 % shorter running time, fuel consumption with turbo-charged wood-gas operation is only 6 % greater over the same route, notwithstanding turbo-charging and the greater engine power. In the case of run *a* the petrol consumption was 30·7 litres per 100 km.

In the case of supercharged engines the charger revolutions normally rise approximately in proportion to the engine output or revolutions, whereas the supercharging pressure increases approximately as the square of the charger revolutions. With a supercharged Diesel engine, where only the combustion air and not the quantity of fuel is supplied in function of the charging pressure, care merely has to be taken that at least the minimum excess of air required for satisfactory combustion is always available, a requirement which is usually likewise fulfilled at low charger revolutions. In the case of supercharged gas operation, however, also the fuel is supplied in relation to the charging pressure. In consequence, with the means hitherto employed the maximum torque could only be attained

line roughly connecting the lower angles of the hatched triangles. At high engine revolutions the charging pressure is kept approximately constant by automatic regulating gear. As a result the engine operates in the speed range between the individual gears under similar conditions to when working non-supercharged, i. e., under an approximately constant, but greater pressure, so that the engine torques are higher, although virtually retaining their original characteristic. A criterion for the success of this process is a high supercharger efficiency, such as is achieved with Brown Boveri exhaust turbo-chargers. If the engine were to be operated with the increased supercharging pressure corresponding to the dash-line portion of the charging pressure curve b in Fig. 4, any type of engine could be supercharged



Fig. 6. — 6-ton lorry with 100 H. P. FBW Diesel engine.

The illustration shows an exhaust turbo-charger fitted under the bonnet of the engine. After conversion to producer-gas operation the same performance was obtained with this vehicle as previously when running on oil.

not only up to, but even beyond its normal output. This possibility is intentionally left unexploited, however, because of the resulting unfavourable torque characteristic and inevitable overloading of the engine which would result.

A special 3·6 litre Opel-Blitz wood-gas-operated test car was employed for extensive investigations out on the road and in the test bay, so as to make sure that the proposed measures would actually produce the desired running characteristic. With this vehicle, equipped with a high-speed engine, it proved possible to increase the power by fully 60 % compared to non-supercharged working, which suffices to give back to this model its full capacity even when loaded to the extreme.

This was corroborated by test, a 4·3 t trailer being attached to the test car which has a tare of 6·8 t, thus giving the — for these conditions — exceptionally heavy total load of 11·1 t. Whereas, without supercharging of the engine the vehicle was incapable of hauling a trailer up 10 % gradients the application of turbo-charging enabled it to climb such hills with ease. No special stress need be laid on the fact that the field of application of vehicles with turbo-charged engines has thus been widely extended, particularly in hilly districts (Fig. 5).

Turbo-charging is also of great value where flat country is concerned. Should there be a short climb on the route to be covered for which the power attainable with producer gas proves inadequate, the load must be correspondingly reduced, although it could easily have been hauled over the other sections of the route. Notwithstanding the fact that turbo-charging is only required for a short period in such a case it is of decisive importance. Not only where producer-gas-operated engines have no reserve of power, however, is turbo-charging indicated: even if ample power is available, such as in the case of vehicles with large FBW (Fig. 6), Saurer, or Berna engines the increase in output enables shorter running times to be achieved.

The development of this patented turbo-charging process permits vehicles to be operated on solid fuels with approximately the same power as on petrol or oil. Interesting prospects are thus opened up for the future. The substantial difference in price between liquid and solid fuels will probably always remain. Due to the fact that exhaust turbo-charging gives back to the producer-gas-operated vehicle its earlier power the initial outlay in connection with the fitting of a charger will be quickly repaid, especially when the speed restrictions due to the shortage of tyres are abolished. Notwithstanding the big drop in road traffic, over 100 plants with an aggregate power of more than 10,000 B. H. P. have already been supplied or are in hand, and with the development of a small air-cooled charger weighing only 18 kg (Fig. 7) we have made another big step forward.

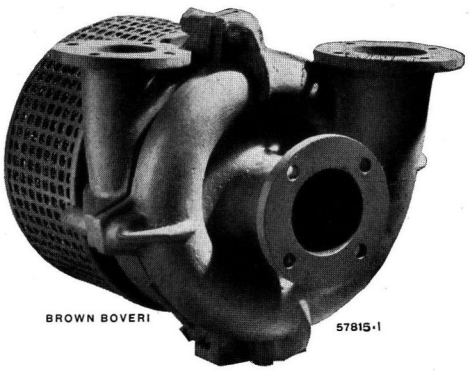


Fig. 7. — Exhaust turbo-charger VTx 95 for producer-gas plants of 70—120 H. P. rating.

In contradistinction to the water-cooled exhaust turbo-charger VTx 110 the turbine casing of this model is air-cooled, so that the engine cooler has not to deal with additional turbine cooling water. The compact design, low weight of only 18 kg, and elimination of cooling water pipes considerably simplifies installation in the vehicle.

(MS 983) *W. Negretti. (E. G. W.)*

A SUCTION-PRESSURE REGULATOR FOR GAS-EXHAUSTERS.

Decimal Index 621.639—53
621—53 : 621.639

Because of the long pipe lines and of the accuracy with which the pressure has to be maintained, the regulation of the suction pressure of gas-exhausters for coke-oven plants and similar installations is often difficult to accomplish. The problem is most simply solved by means of a small suction pressure regulator with independently adjustable temporary and permanent static drops, so that by making the temporary drop sufficiently large, stability is ensured, while the required accuracy of pressure regulation is achieved by making the permanent drop small.

THE exhaustion of gas from coke-oven plants is subject to the difficult condition that the pressure in the distilling retorts must coincide with the atmospheric pressure to within 1 mm w. g. or less. The reason for this unusual requirement lies in the porous nature of the retort walls which upon the occurrence of a vacuum, allow combustion gases from the surrounding ducts to be drawn into the retort and to contaminate the coke-oven gas, or, upon occurrence of a rise in pressure allow valuable gas to escape to the atmosphere. These strict requirements in regard to accuracy present a problem for the regulator, which can be solved only by subdividing the functions of the latter into coarse and fine regulation. Coke-oven regulators are therefore built according to the fundamental principle illustrated in Fig. 1, in which the subdivision of the functions can be clearly seen.

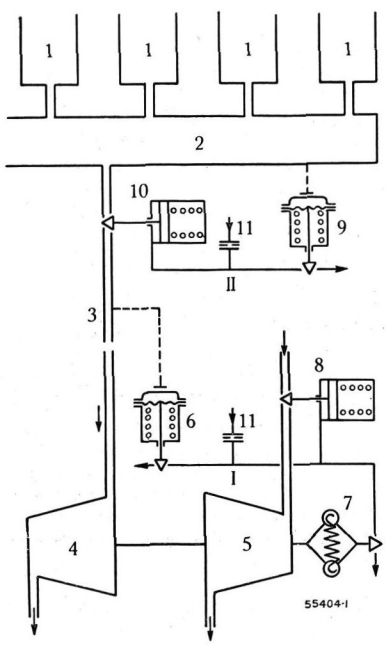


Fig. 1. — Diagram of gas-exhauster regulation for a coke-oven plant.

The subdivision into fine regulation (at the coke-oven battery) and coarse regulation (at the gas-exhauster) usually resorted to is clearly visible.

1. Coke-oven battery.
2. Header.
3. Suction pipe.
4. Gas-exhauster.
5. Turbine drive.
6. Coarse pressure regulator.
7. Speed regulator (as speed limiter).
8. Steam inlet valve.
9. Fine pressure regulator.
10. Throttling device.
11. Baffle plate for control oil supply.

I. Oil pressure system of coarse regulator.
II. Oil pressure system of fine regulator.

The coarse regulation keeps the pressure at the end of the suction pipe approximately constant, by causing the suction pressure regulator 6 to vary the speed of the gas-exhauster accordingly. It achieves this by varying the position of the steam inlet valve 8 of the driving turbine 5, thereby affecting the driving power of the exhauster and hence its speed. The maintenance of an approximately constant pressure at the end of the suction pipe provides the condition for the proper operation of the fine regulation, the function of which is to keep constant with the required degree of accuracy, the pressure in the header-pipe 2. The fine regulation works with an extremely sensitive regulator 9, which by means of the throttling device 10, adjusts the flow of the exhausted gas according to the quantity of gas generated in the coke-oven battery.

Whereas the fine regulation is mainly a question of obtaining the necessary high sensitivity, the chief difficulty of the coarse regulation lies in ensuring proper stability. The usually long pipe lines between the headers and the gas-exhausters necessitate, for stable operation, such a large static drop that the resulting pressure variations after the throttling valve 10 of the fine regulation exceed by far the maximum admissible values. The problem of regulating the pressure at the end of the long suction line with some degree of constancy can therefore be solved only with regulators having two independent characteristics, a temporary one and a permanent one, each capable of being adjusted independently of the other. Such a regulator offers the possibility of making the temporary static drop sufficiently large to ensure stable operation, and the permanent drop as small as is required to achieve the desired accuracy of pressure regulation.

Fig. 2 shows the principle of the suction pressure regulator developed by Brown Boveri for such plants. Its main feature is not so much its high sensitivity as the large range of adjustability of the temporary and permanent static drops.

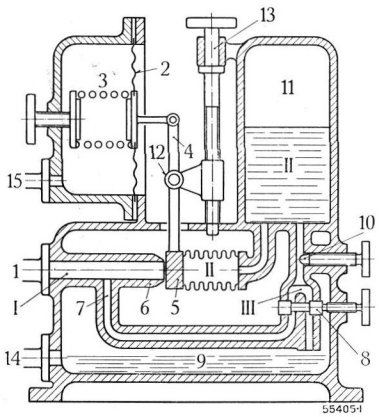


Fig. 2. — Diagram of the suction pressure regulator for oil pressure controlled turbine gas-exhausters with independently adjustable temporary and permanent static drops.

1. Connection to turbine governing oil system.
2. Suction pressure diaphragm.
3. Regulator spring.
4. Lever.
5. Impact plate.
6. Governing oil outlet.
7. By-pass to 8.
8. Throttling device.
9. Oil chamber.
10. Throttling device.
11. Air cushion.
12. Adjustable fulcrum.
13. Adjusting screw for 12.
14. Oil outlet.
15. To suction pipe.

I, II and III: Oil systems.

The regulator is connected by means of the pipe to the oil-pressure system of the turbine driving the gas-exhauster where it controls the inlet valves in parallel with the speed regulator. The latter is adjusted so that it comes into operation only when the speed tends to rise to an excessive value. The suction pressure regulator receives its impulse from the suction pipe after the throttling device of the fine regulator. The force developed by suction on the diaphragm 2 is transmitted partly to a spring 3 and partly to the impact plate 5 through the lever 4. The impact plate is the real control organ of the regulator. It builds up the pressure in the system I to a certain value which is proportional to the force with which the plate is pressed against the mouth of the escape tube 6 of this system. This force determines, therefore, the opening of the inlet valves on the turbine and hence its output. It is, however, operated not only by the force transmitted by the lever 4 to the impact plate, but also by the force due to a second oil pressure system II acting through a metallic diaphragm having an important effect on the equilibrium conditions of the impact plate. The pressure of this system is established as follows: A continuous stream of oil 7 flows through a double throttling device 8 to the pressureless oil chamber 9 from system I. In this manner, a new pressure stage III is created between the system I and that at atmospheric pressure which divides the pressure difference in a constant ratio, but which can be adjusted by the throttling device 8. This pressure is then transmitted through a further throttling device 10 to the system II, but always after the lapse of a deliberatly introduced time delay which depends on the size of the opening 10 and the volume of the air cushion 11. This time delay ensures that the regulator is affected only by the permanent and not by the temporary conditions. The temporary static drop is affected, therefore, only by the force transmitted to the impact plate 5 by the diaphragm 2 and which depends on the position of the fulcrum 12 on the lever 4. The screw 13 enables the point of rotation of the lever to be moved in a vertical direction and thereby the temporary static drop to be varied. The range of adjustment is about 1 : 10.

The permanent static drop on the other hand depends mainly on the ratio of the pressures in the systems I and II, that is, on the setting of the throttling device 8. This device enables the permanent static drop to be varied over an amount ranging from the full value of the temporary static drop down through zero to negative values. The possibility of using a negative static drop has been provided for those cases in which it is desired to assist the action of

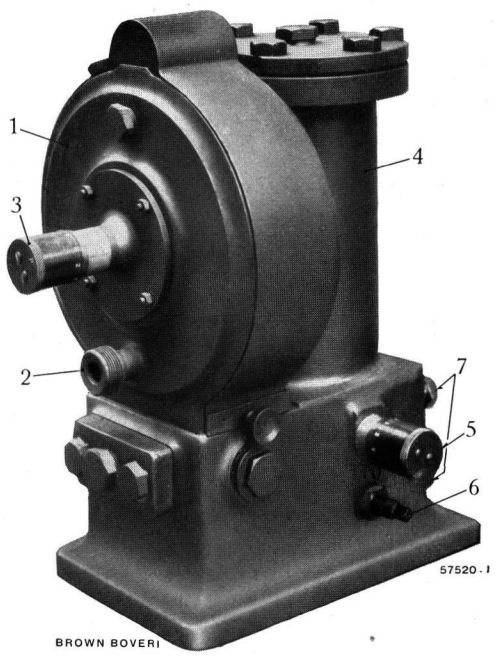


Fig. 3. — Suction pressure regulator according to Fig. 2.

The insensitivity is about 1 mm w. g. The suction pressure can be adjusted within a range of 0—500 w. g.

1. Diaphragm casing.
2. Connection for the impulse pressure.
3. Adjusting screw for pressure setting.
4. Air cushion.
5. Adjustment for temporary static drop.
6. Adjustment for time delay between temporary and permanent conditions.
7. Connections for control of oil and oil outlet.

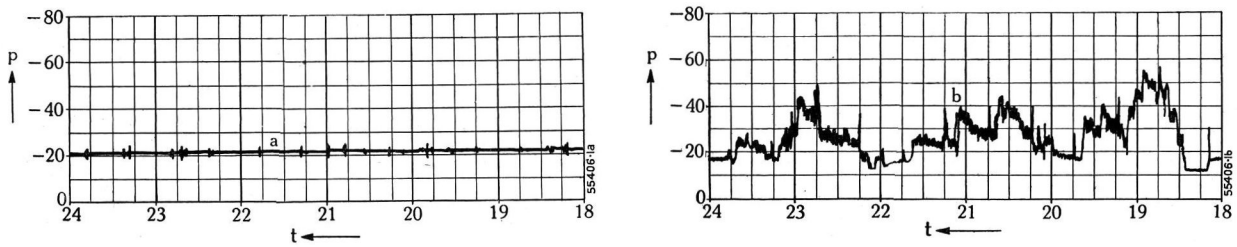


Fig. 4. — Variation of the pressure in the gas suction pipe of the gas-exhauster set at the Basle Municipal Gasworks.

a. With the new regulator. t. Time in hours.
b. With the old regulator. p. Vacuum in mm w. g.

Except for the momentary peaks due to the large temporary static drop, the new regulator keeps the pressure accurately constant to 1 mm w. g.

a fine regulator by increasing the pressure drop in the fine regulator control device with increasing flow and to reduce it with decreasing flow.

The time delay between temporary and permanent conditions is adjusted by means of the screw 10, whereas the value at which the suction pressure is required to be maintained is adjusted by varying the compression of the spring 3. It is evident from the manner in which these adjustments are effected that the settings can be varied continuously and without jumps.

This regulator is a success in every respect. The first model, shown in Fig. 3, was supplied to the Basle Municipal Gasworks where it replaced a regulator of an older type. — The excellent regulating characteristics can be seen from the diagram Fig. 4 a, which shows the behaviour of the pressure in the gas suction pipe over a period of several hours. The advantages obtained can be even better appreciated if this diagram is compared with the diagram Fig. 4 b of the old regulator. The fine peaks in the curve of the new regulator indicate the large temporary static drop which has to be allowed to ensure stability. The variations of pressure fluctuations are, however, only of short duration and the pressure always returns to the same value, the variations of which are hardly noticeable, affording a proof of the high sensitivity characteristic of the regulator.

The good results obtained have already induced the Basle Gasworks to place an order for a second regulator intended for a further gas-exhauster set.

(MS 937) *A. Leyer. (Hv.)*

EARTH-FAULT PROTECTION WITH ARC SUPPRESSION COILS. THE AUTOMATIC ADJUSTMENT OF THE COIL CURRENT.

Decimal Index 621.316.935

The present article deals with the automatic adjustment of the current of arc suppression coils in relation to varying network conditions. The problem of the automatic control of a single adjustable coil and the parallel operation of several control gears with position correction of the adjusting gear can now be considered as completely solved. This obviates the continual supervision of the earth-fault protection gear by the operating staff, a requirement which is being increasingly specified at the present time. It is therefore to be assumed that the progressively adjustable arc suppression coil with automatic current adjustment will prove popular in power distribution circles.

I. ADVANTAGES OF ADJUSTABLE ARC SUPPRESSION COIL.

THE big advantages accruing from the earth-fault protection of overhead transmission and cable systems by arc suppression coils connected to the system neutral are now generally recognized. These are briefly as follows:—

(a) Rapid extinction of the fault arc.

(b) Obviation of intermittent earth faults through retardation of the recovery voltage at the point of fault, thus preventing re-ignition of the arc. (Over-voltages produced by intermittent earth faults are some of the severest encountered and are thus greatly apprehended.)

(c) Protection of generators through compensation of system capacitive load (capacitance of three system phases to earth), so that no capacitive load surges are produced by an earth fault. In certain cases this also improves stability.

(d) The suppression of the earth-fault current at the point of fault.

Although these advantages will not be gone into here a very important point should be remembered. Since even the most modern arrestors with their high discharge capacity are incapable of sustaining the over-voltages resulting from earth faults, their operating

voltage must be selected much higher than when arc suppression coils are employed for the protection of a system. As a matter of course, however, the raising of the level of protection affects the other levels in the same sense. Even only on this count the arc suppression coil method of earth-fault protection forms an extremely economic solution.

The foregoing advantages of the arc suppression coil have contributed to its wide application. At first sight, therefore, it seems astonishing that a whole number of undertakings should still be operating without some kind of earth-fault protection. Upon closer investigation of this paradox, however, there is found to be a good reason for the reluctance of electricity supply undertakings to adopt the arc suppression coil for earth-fault protection purposes, viz., the difficulty of correctly adjusting the coil under service conditions. In order rightly to evaluate this difficulty it should be remembered that full earth-fault protection is only obtained when the arc suppression coils are so adjusted that the total coil current is practically of the same magnitude as the earth-fault current of the system under consideration. Every time the earth-fault current is increased or reduced through the switching in or out of lines the coil current must also be adjusted to a corresponding higher or lower value. This presupposes that the operating engineers are always aware of the magnitude of the earth-fault current and, moreover, are kept informed of system conditions. In extensive, interconnected, medium-voltage systems, in particular, such details — if not altogether impossible to obtain — involve complicated equipment and extra work. What is more, it is generally at the very time when the operating engineers are fully occupied due to load changes in the system that this additional work also has to be done.

Another difficulty is that the majority of arc suppression coils have a built-in off-load tap-changing switch, i.e., the coil must be disconnected from the system to adjust the current and must be switched in again after adjustment. The change-over is thus cumbersome and far from meets the simplicity and reliability demanded of modern installations. For this reason arc suppression coils have been increasingly fitted with built-in on-load tap-changing switches for altering the current setting. Although this solves the current adjustment problem coils with on-load switches are not looked upon with universal favour due to the limited range of regulation and restricted number of tappings.

It was the development of the Brown Boveri arc suppression coil with progressive current adjustment [1] which again drew attention to the desirability of so simplifying the adaptation of the earth-fault protective equipment to system conditions that operating engineers would be relieved of all supervision and re-adjustment work. This obviously called for full-automatic gear.

With the realization of the automatic current adjustment of arc suppression coils in relation to the expansion of the system a really ideal method of earth-fault protection has been evolved which in conjunction with the progressively adjustable arc suppression coil gives optimum advantages, viz.:—

(a) The operating engineers are completely relieved of all supervision work.

(b) The progressive current adjustment permits of accurate adaptation of the coil to the prevailing conditions.

(c) Due to the wide range of regulation of the progressively adjustable coil (1 : 8 or in certain cases 1 : 10) only a restricted number of adjustable coils is necessary even in extensive systems.

Hereafter the design and mode of operation of the automatic control gear are described.

In a system with automatically controlled arc suppression coils the operating engineers have merely to see that there is always sufficent coil capacity available in the system, i. e., for all possible network switching conditions. When planning earth-fault protection, therefore, the simplest method is to split up the system into a number of sections (preferably sections which can be operated independently) and to allot to each section a number of coils of which one should have an ample range of regulation. If a section of the system is cut in or out the corresponding coils will then be switched at the same time, whereas the disconnection of individual lines or small sections of systems involves re-setting of adjustable coils. If a system is not very extensive a single adjustable coil will suffice and the automatic control will operate solo. In large systems, however, several adjustable coils may be provided, each with its own automatic gear and the various control gears will have to operate in parallel. The two cases will be dealt with separately hereafter.

[1] For the description of arc suppression coils with progressive current adjustment see the Brown Boveri Review 1943, page 103.

Even in systems already protected by arc suppression coils the earth-fault protective equipment can be modernized by installing a progressively adjustable coil with a sufficiently wide range of regulation and automatic current adjusting gear.

II. AUTOMATIC CONTROL GEAR IN SOLO OPERATION.

(a) *Principle of Control.*

Fig. 1 shows the principle of the automatic control gear in solo operation. An auxiliary voltage U is connected in series with an arc suppression coil. The latter, together with the transformer, the system capacitance to earth, and the earth itself form the operating circuit in which a current J flows.

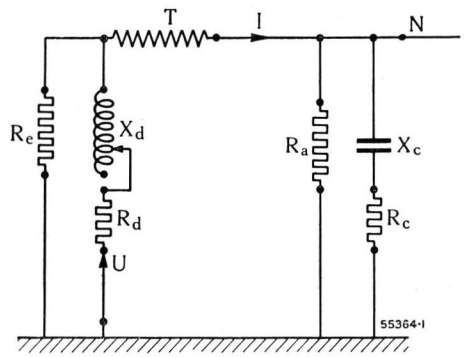


Fig. 1. — Simplified diagram for automatic control of arc suppression coils.

U. Operating voltage.
J. Operating current.
N. System.
T. Transformer.
R_e, R_d, R_a, R_c } Loss resistance of coil or system.
X_c. Reactance of system to earth.
X_d. Reactance of arc suppression coil.

The losses in the arc suppression coil are split up into voltage and current dependent losses, i. e., core and winding losses, by a parallel resistor R_e and a series resistance R_d. The variable reactance of the coil is represented by X_d. On the network side the three windings of the transformer T and the three phases of the system N are taken collectively. Here the system capacitance to earth is symbolized by the reactance X_c with loss R_c. Finally, the resistance R_a stands for the losses in the leads. When further coils are connected to the system their reactances and loss resistances are represented by X_c, R_c, and R_a.

To facilitate calculation the single-pole simplified diagram in Fig. 1 is replaced by the single-pole equivalent diagram in Fig. 2, whereby:

$$R_1 = \frac{R_e\{R_d(R_e+R_d)+X_d^2\}}{(R_e+R_d)^2+X_d^2} \quad (1a)$$

$$X_1 = \frac{R_e^2 X_d}{(R_e+R_d)^2+X_d^2} \quad (1b)$$

and

$$R = \frac{R_a\{R_c(R_a+R_c)+X_c^2\}}{(R_a+R_c)^2+X_c^2} \quad (2a)$$

$$X = \frac{R_a^2 X_c}{(R_a+R_c)^2+X_c^2} \quad (2b)$$

If it is assumed that R_e is large and R_d small compared to X_d and R_a and R_c are also large and small,

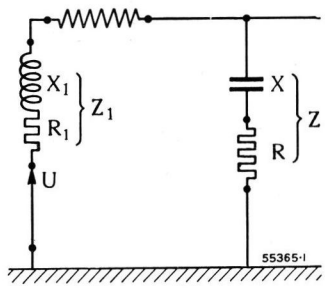


Fig. 2. — Equivalent diagram of automatic control of arc suppression coils.
(For denotation see text.)

respectively, compared to X_c formulæ (1a) to (2b) can be simplified as follows with a very good approximation.

$$R_1 = R_d + \frac{X_d^2}{R_e} \quad (3a)$$

$$X_1 = X_d \quad (3b)$$

$$R = R_c + \frac{X_c^2}{R_a} \quad (4a)$$

$$X = X_c \quad (4b)$$

Due to the auxiliary voltage U current J flows in the operating circuit as per the formula:

$$J = \frac{U}{Z+Z_1} \quad (5)$$

where Z is the system impedance to earth and Z_1 the impedance of the arc suppression coil. Now

$$Z = R + jX \quad (6a)$$

$$Z_1 = R_1 + jX_1 \quad (6b)$$

and if the expressions for Z and Z_1 are inserted in the equation (5) for the operating current the result will be:

$$J = \frac{U}{R+R_1+j(X+X_1)} \quad (7)$$

The operating current has an active component J_w and a reactive component J_{bl}, inasmuch as from equation (7) it results that

$$J = U\frac{R+R_1-j(X+X_1)}{(R+R_1)^2+(X+X_1)^2} \tag{8a}$$

$$J_w = U\frac{R+R_1}{(R+R_1)^2+(X+X_1)^2} \tag{8b}$$

$$J_{bl} = U\frac{X+X_1}{(R+R_1)^2+(X+X_1)^2} \tag{8c}$$

If the current setting of the arc suppression coil is varied, but all other essentials are left unchanged the current J in the operating circuit is altered. To enable a good idea of these modifications to be obtained both the active and the reactive components of the operating current will be considered more closely. When the values for R, X, R_1, and X_1 from formulæ (3a) to (4b) are inserted

$$J_w = U\frac{R_d+R_c+\frac{X_d^2}{R_e}+\frac{X_c^2}{R_a}}{\left(R_d+R_c+\frac{X_d^2}{R_e}+\frac{X_c^2}{R_a}\right)^2+(X_d+X_c)^2} \tag{9a}$$

and

$$J_{bl} = U\frac{X_d+X_c}{\left(R_d+R_c+\frac{X_d^2}{R_e}+\frac{X_c^2}{R_a}\right)^2+(X_d+X_c)^2} \tag{9b}$$

The only variable in these two formulæ is the reactance X_d of the arc suppression coil. Two very important points must first be observed:

$$\text{for } X_d = -X_c \begin{cases} \text{(a)} \ldots J_{bl} = 0 \\ \text{(b)} \ldots J_w = U\dfrac{1}{R_d+R_c+\frac{X_d^2}{R_e}+\frac{X_c^2}{R_a}} \end{cases}$$

The adjustment of the coil given by $X_d = -X_c$ is the resonance setting.

This is characterized by the zero reactive component of the current in the operating circuit, the purely non-inductive operating current, and the maximum value of the active component. Nevertheless, this maximum value is not absolute, but is dependent on the loss resistances of both the system and the coil to a high degree. For this reason the active component cannot be employed for the supervision of the adjustment of the coil. On the other hand, the reactive component is particularly suitable for this purpose, since its zero value is entirely independent of all loss resistances.

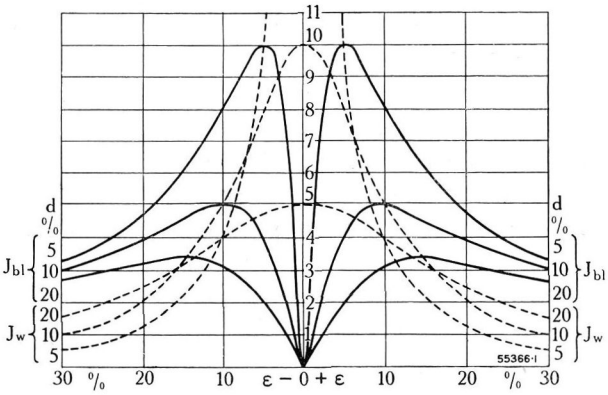


Fig. 3. — Active and reactive components of operating current plotted in function of the per cent de-tuning ε of the arc suppression coil with different ratios *d* between reactance and non-inductive resistance.

J_w. Active component.
J_{bl}. Reactive component.

Since, independent of the ratio between the reactance and the non-inductive resistance, the reactive component is zero with resonance tuning of the arc suppression coil, this component can be used for the resonance adjustment of the coil.

The curves in Fig. 3 illustrate the conditions in the operating circuit, viz., the active component J_w and the reactive component J_{b1} of the operating current for different losses in the coil and system. The ratio between the reactance and the non-inductive resistance in per cent is given at d. The curves are also plotted in relation to the percent detuning of the coil which is given by $\varepsilon = \dfrac{X_c - X_d}{X_d}100$.

From these curves it is obvious that the resonance setting ($\varepsilon = 0$) is positively determined by the zero value of the reactive component of the operating current and not by the magnitude of the active component.

It is moreover interesting to find that both for earth-fault protection, where the arc suppression coil is connected in parallel with the earth capacitance of the system and in the operating circuit where the coil and the system capacitance are in series, the resonance setting of the coil is characterized by exactly the same relation $X_d + X_c = 0$. In consequence, the resonance setting of the arc suppression coil determined from $J_{bl} = 0$ can be transferred directly from the operating circuit to the earth-fault protective gear.

(b) *Effect of System Asymmetry on Operating Circuit.*

In a system having equal values of capacitance for each of the three phases to earth there is no current through the coil under normal conditions and the reactive component of the operating current can

be employed both to determine the adjustment of the coil and for automatic control purposes.

In asymmetrical systems, however, a current, due to the system asymmetry to earth, flows through the coil under normal service conditions and interferes with the operating circuit. The magnitude and direction of the asymmetry current are given chiefly by the asymmetry voltage V_o of the system neutral point to earth, whereby all arc suppression coils are disconnected. This is (Fig. 4)

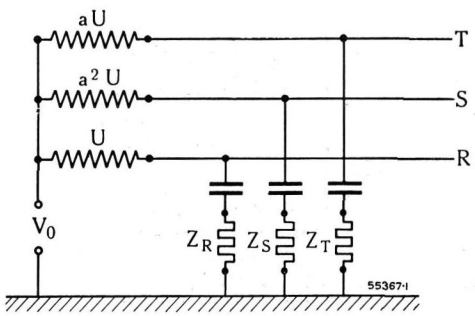


Fig. 4. — Diagram for determining the asymmetry voltage of a three-phase system.

U, a^2U, $a\,U$. System of phase voltages.
V_o. Asymmetry voltage.
Z_R, Z_S, Z_T. Impedances of three system phases R, S, and T to earth.

$$V_o = -\frac{\frac{U}{Z_R} + a^2\frac{U}{Z_S} + a\frac{U}{Z_T}}{\frac{1}{Z_R} + \frac{1}{Z_S} + \frac{1}{Z_T}} \qquad (10)$$

In this formula

U, a^2U, aU denote the system of three phase-voltages

Z_R, Z_S, Z_T denote the system of the three equivalent impedances of the phases to earth from

$$\frac{1}{Z_R} = \frac{1}{R_{a_R}} + \frac{1}{R_{c_R} + jX_{c_R}} \quad \text{etc.}$$

After connection of the arc suppression coils there flows in the system the asymmetry current

$$J_o = \frac{V_o}{Z + Z_1} \qquad (11)$$

The total current in the operating circuit is thus [from formulæ (5) and (11)]

$$J + J_o = \frac{U + V_o}{Z + Z_1} \qquad (12)$$

The reactive component of the total current J_{tot} could also be used for operation and automatic control if it were possible to employ the voltage $U + V_o$ as supply voltage. This, however, is out of the question, for the voltage V_o can only be measured as long as no arc suppression coils are connected to the system.

The fact that the operating voltage and the asymmetry current do not form a power product if they are of different frequency, can be exploited here. Since the asymmetry current is at power frequency another frequency is selected for the operating voltage and the reactive power measured. When

$u = U_m \sin \omega_1 t \qquad \omega_1 = 2\pi f_1$

$i_{bl} = J_{bl_m} \sin (\omega_1 t - \varphi_1) \quad \omega = 2\pi f$

$i_o = J_{o_m} \sin (\omega t - \varphi) \quad f$ = Power frequency

f_1 = Operating frequency

the product

$$\begin{aligned} u i_{bl} &= U_m J_{bl_m} \sin \omega_1 t \sin (\omega_1 t - \varphi_1) \\ &= \tfrac{1}{2} U_m J_{bl_m} \{\cos \varphi_1 - \cos (2\omega_1 t - \varphi_1)\} \qquad (13) \\ &= U J_{bl} \cos \varphi_1 - U J_{bl} \cos (2\omega_1 t - \varphi_1) \end{aligned}$$

The mean value of this reactive power is $UJ_{bl} \cos \varphi_1$. In addition, pulsations with twice the frequency occur. The product

$$\begin{aligned} u i_o &= U_m J_{o_m} \sin \omega_1 t \sin (\omega t - \varphi) \qquad (14) \\ &= \tfrac{1}{2} U_m J_{o_m} [\cos \{(\omega_1 - \omega) t + \varphi\} - \cos \{(\omega_1 + \omega) t - \varphi\}] \\ &= U J_o \cos \{(\omega_1 - \omega) t + \varphi\} - U J_o \cos \{(\omega_1 + \omega) t - \varphi\} \end{aligned}$$

The mean value of this power is zero. Two pulsations are set up, one with a frequency of $f_1 - f$ and the second with a frequency of $f_1 + f$. Since these rapid pulsations do not affect the relay, due to the inertia of its motive system, the powers are as follows:—

Operating power $UJ_{bl} \cos \varphi_1$

Interference power $UJ_o \cos \{(\omega_1 - \omega) t + \varphi\}$

Care must be taken to choose the frequency f_1 to give a difference of $f_1 - f$ large enough to prevent the relay following the pulsation $UJ_o \cos \{(\omega_1 - \omega)\, t + \varphi\}$. On the other hand, the difference $f_1 - f$ must not be selected inordinately large otherwise the resonance setting of the arc suppression coil at the operating frequency will differ excessively from that at the power frequency. A strong damping device in the relay is chiefly instrumental in keeping the difference between the operating and power frequencies small.

(c) *Connection Details.*

As will be seen from the diagram in Fig. 5 the operating voltage U and operating current J are supplied by a frequency modulator 2 the primary of which is connected to a system with the power frequency f and its secondary in the lead between the arc suppression coil and the earth through an

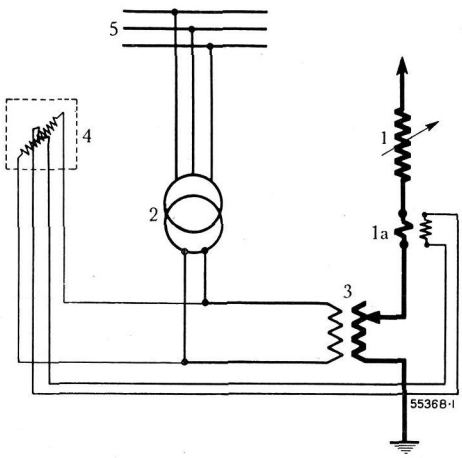


Fig. 5. — Connection of control relays.

1. Adjustable arc suppression coil.
1a. Built-in current transformer.
2. Frequency modulator.
3. Auxiliary transformer.
4. Control relay.
5. System.

The frequency modulator supplies the operating voltage and current through transformers to the control relay.

auxiliary transformer 3. The voltage U can be adapted to the system conditions by means of tappings on the auxiliary transformer. The operating frequency f_1 can also be varied on the modulator.

Reactive power relays 4 are employed to control the arc suppression coil, these being connected to the voltage U of the modulator either directly or through a transformer. They also measure the operating current J and for this purpose are connected to a transformer 1a generally incorporated in the arc suppression coil. The contact of one relay closes when the coil is set to too low a current and causes the current setting to be increased. The contact of the other relay closes when the adjusted current is excessive and has to be regulated by the control gear to a lower value. Although the heavily damped moving system precludes hunting to a great extent the contacts must be designed for rapid and reliable closing and must not continually open and close. They are therefore of the magnetic snap-action type, a design which has been employed with success in the quick-acting regulator field for many years past.

Fig. 6 depicts a control relay as specially developed for the automatic control of adjustable arc suppression coils. The relay is housed in a standard casing, size 1, for flush mounting in a switchboard. In the case of coils with current adjustment by on-load tap-changing switch in particular it is important that the operating phase angle of the control relay be adjustable. Since this is fixed, however, transformers with tappings, in series with the high-resistance (voltage) coils are necessary for adjustment purposes (Fig. 7). That it is imperative for the operating phase angle to be adjustable can best be proved by means of the vector diagram for the operating current J. From formula (7) the operating current is given by

$$J = \frac{U}{R + R_1 + j(X + X_1)}$$

This presupposes that the system conditions remain constant, while the coil is being re-set, i. e., the reac-

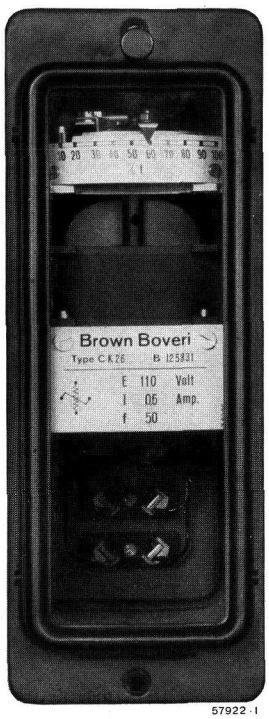


Fig. 6. — Control relay type CK 26, specially designed as heavily-damped reactive-power relay without reverse torque for the automatic control of arc suppression coils.

tance X_1 altered. If as a first approximation it is assumed that the losses in the coil are not affected by the setting ($R_1 =$ constant) the vector diagram of the operating current J becomes a circle, as shown in Fig. 8. (The circle is found by inversion of the straight line "X_1 = variable"; diameter of circle $\frac{U}{R + R_1}$). In the case of current adjustment by means of an on-load tap-changing switch the reactance X_1 cannot be progressively regulated, so that the individual values of the reactances X_1, X_2, etc. in the vector diagram correspond to the operating currents J_1, J_2, etc. The operating phase angles of the control relays must now be selected so that there is only one single step of the arc suppression coil between them. If the

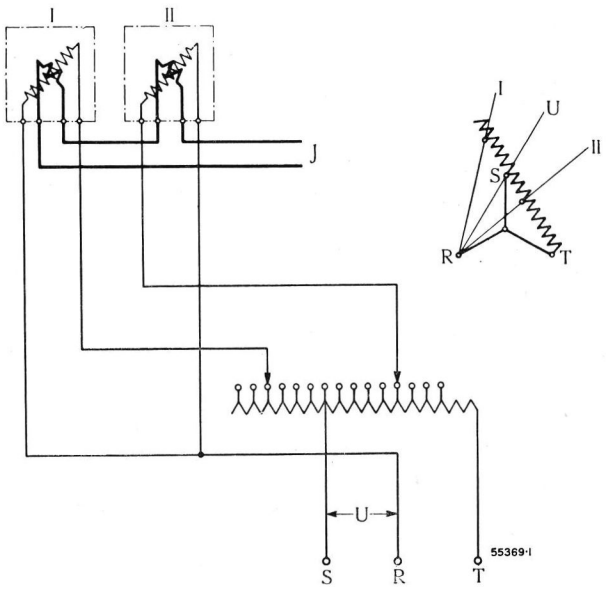


Fig. 7. — Adjustment of operating phase angle on tapping transformer in voltage circuit of relays. The possibility of adjustment of the operating phase angle on the control relays enables the control to be used for the most widely differing types of arc suppression coils.

operating phase angles had no step between them continual "pumping" of the control gear would result and the on-load tap-changing switch would be constantly moved backwards and forwards. On the other hand, there must not be more than one step between the operating phase angles, otherwise it would be impossible for the switch to function in this zone. For these reasons an arrangement was selected with which the operating phase angle of the control relays can be rotated within very wide limits and so enable them to be adapted to different types of arc suppression coil

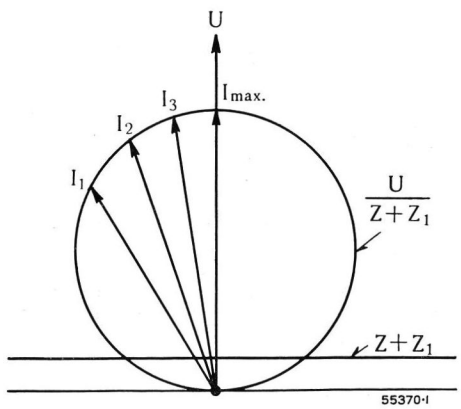


Fig. 8. — Approximate diagram for the operating current when adjustment of arc suppression coil varied.

U. Operating voltage.
$J_1, J_2, J_3 \cdots J_{max}$. Operating currents in the various positions of the coil.
$Z+Z_1$ Impedance.
$\frac{U}{Z+Z_1}$ Operating current circle.

(with coarse and fine tappings or progressive adjustment).

From formula (3a) for the loss resistance R_1 it follows that this resistance varies with the setting of the arc suppression coil. The vector diagram thus deviates from the circle, but only slightly and is therefore not taken into further consideration. Fig. 9 is the circle diagram for a coil regulated in steps. The operating phase angles of the two relays are so selected that there is one arc suppression coil step between them.

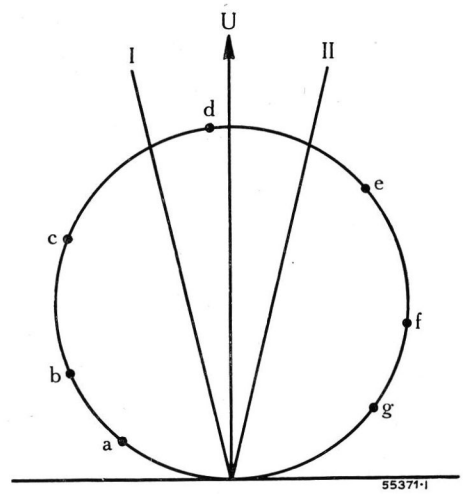


Fig. 9. — Arc suppression coil with current adjustment by on-load tap-changing switch. The points a—g correspond to the different positions of the on-load tap-changing switch.

U. Operating voltage.
I, II. Operating phase angles of the two control relays.

III. AUTOMATIC CONTROL GEARS IN PARALLEL OPERATION.

This problem arises when automatically controlled arc suppression coils are provided at two different points in a system (see Fig. 10) and the two coils have to operate in parallel.

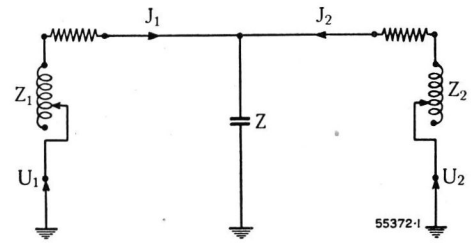


Fig. 10. — Simplified diagram of automatic controls in parallel operation.

U_1, U_2. Operating voltages.
J_1, J_2. Operating currents.
Z. Impedance of system to earth.
Z_1, Z_2. Impedances of arc suppression coils.

The first question is that of the distribution of the control currents J_1 and J_2 throughout the system, it being assumed that the frequency modulators (with voltages U_1 and U_2) both operate with the frequency f_1. Let the impedances of the adjustable coils be Z_1 and Z_2, that of the system, including any further coils, Z (impedance of three phases to earth in parallel connection).

For this connection

$$U_1 - J_1 Z_1 - (J_1 + J_2) Z = 0 \qquad (15a)$$

$$U_2 - J_2 Z_2 - (J_1 + J_2) Z = 0 \qquad (15b)$$

From these two equations

$$J_1 = \frac{U_1}{Z_1} - \frac{\frac{U_1}{Z_1} + \frac{U_2}{Z_2}}{\left(\frac{1}{Z} + \frac{1}{Z_1} + \frac{1}{Z_2}\right) Z_1} \qquad (16a)$$

$$J_2 = \frac{U_2}{Z_2} - \frac{\frac{U_1}{Z_1} + \frac{U_2}{Z_2}}{\left(\frac{1}{Z} + \frac{1}{Z_1} + \frac{1}{Z_2}\right) Z_2} \qquad (16b)$$

Provided that the arc suppression coils operate close to the resonance setting the factor $\frac{1}{Z} + \frac{1}{Z_1} + \frac{1}{Z_2}$ will be relatively small in relation to the factor $\frac{1}{Z_1}$ or $\frac{1}{Z_2}$. Thus as an approximation

$$J_1 = \frac{\frac{U_1}{Z_1} + \frac{U_2}{Z_2}}{\left(\frac{1}{Z} + \frac{1}{Z_1} + \frac{1}{Z_2}\right) Z_1} \qquad (17a)$$

and

$$J_2 = \frac{\frac{U_1}{Z_1} + \frac{U_2}{Z_2}}{\left(\frac{1}{Z} + \frac{1}{Z_1} + \frac{1}{Z_2}\right) Z_2} \qquad (17b)$$

The operating currents J_1 and J_2 are greatest when the operating voltages of the frequency modulators have the same phase displacement. This fact is exploited for the connection of the modulators in parallel.

If the modulators operate in parallel with the correct phase displacement and their voltages are selected of the same magnitude, so that

$$U_1 = U_2 = U$$

then upon inserting this value in the equations (16a) and (16b)

$$J_1 = \frac{U}{Z_1} \frac{\frac{1}{Z}}{\frac{1}{Z} + \frac{1}{Z_1} + \frac{1}{Z_2}} \qquad (18a)$$

$$J_2 = \frac{U}{Z_2} \frac{\frac{1}{Z}}{\frac{1}{Z} + \frac{1}{Z_1} + \frac{1}{Z_2}} \qquad (18b)$$

The operating currents are distributed over the system in the inverse ratio to the impedances of the corresponding coils; they are therefore proportional to the rated currents or outputs of the coils. If identical coils are installed at different points and the adjusted currents are also the same, then $Z_1 = Z_2$ and $J_1 = J_2$. In this case each control gear behaves as though it were in solo operation, only an exactly similar regulating speed being needed to ensure the two coils correctly operating in parallel.

Frequently, however, coils of dissimilar rating are installed at different points and, what is more, the control gears may not have the same operating time. The difference in rating can be compensated for in a simple manner by suitably selecting the transformation ratio of the current transformers supplying the control relays. To ensure all of the coils being uniformly and simultaneously regulated, independent of the regulating speed, and to prevent, for instance, one coil being in the J_{max} position and the other in the J_{min} position, the control gear is also influenced by the setting of the arc suppression coil (position of on-load tap-changing switch or movable core). If a coil is in the wrong position it can be corrected by varying the operating phase angle of the control relay, so that a reactive power is simulated and the arc suppression coil adjusted in the correct sense. Such a correction will usually also be necessary when a coil has been switched in.

The effect of the position correcting gear is described hereafter with the help of a number of diagrams. Let it be assumed that two progressively adjustable coils operate in the vicinity of the resonance setting at points P_1 and P_2 (see Figs. 11a and 11b), but with relatively widely differing adjustments. Let the current of coil 1 be too high and that of coil 2 too low. When both coils are displaced simultaneously the operating currents describe two circles of different diameter, i. e., starting from points P_1 and P_2, to the left when set to a lower current and to the right when adjusted to a higher current.

Without position correction the widely differing adjustment of the two arc suppression coils is readily possible, for both points P_1 and P_2 lie in the working zone between the operating phase angles I and II of the control relays. Position correction is effected by varying the operating phase angles of the control relays.

Beginning with the centre position of the coils the operating phase angle of the control relays of coil 1 is shifted to the right, that of coil 2 to the left.

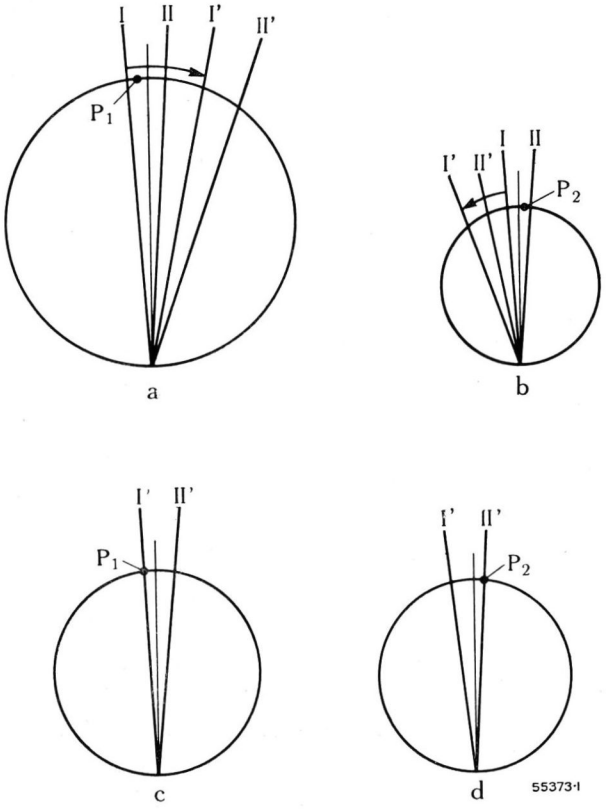


Fig. 11. — **Diagrams of operating currents of two controls in parallel operation with arc suppression coils in widely differing positions.**

I, II. Operating phase angles of control relays without position correction.

I', II'. Operating phase angles of control relays with position correction.

P_1, P_2. Operating points with coils in widely differing positions.

Diagram a. Operating current excessive. The adjusted current of the coil in question is too high. The phase displacement of the control relays must be varied to the right.

Diagram b. Operating current too low. The phase displacement of the control relays must be varied in the other direction. Through the variation of the phase displacement of the control relays regulation with position correction is obtained.

Diagrams c and d. The aim of regulation is achieved; the operating currents coincide with operating points P_1 and P_2 and are virtually of the same magnitude. Both coils are therefore practically in the same position.

Since points P_1 and P_2 now lie outside the dead zone between the control relays both coils are readjusted and move to the centre position. As a result of this movement the variation of the operating phase angles of the control relays will be practically nullified. The final state of regulation is given by the diagrams in Figs. 11c and d. By keeping the angle between the operating phase displacements I and II small the coils can be made to regulate to virtually the same position.

The variation of the operating phase angle of the control relays in relation to the setting of the arc suppression coils is most simply effected by an induction regulator in the voltage circuit of the relays, this being directly operated mechanically by the moving component of the coil, i. e., either the on-load tap-changing switch or the moving core. Since the power consumption of the relay voltage circuit is very low the induction regulator need only be small and, together with the control apparatus for remote position indication, can be built on the arc suppression coil without taking up much additional space.

IV. CONCLUSION.

The entire automatic control gear is designed to take account of all practical requirements. It can be changed over from automatic to remote push-button control or can be cut out in a simple manner; in the latter case only the mechanical hand and electrical remote control gear is left in operation.

In the event of an earth fault the automatic gear is blocked, so that the arc suppression coil cannot be re-adjusted as long as the fault lasts; immediately the fault is cleared, however, it is released.

The control gear needs very little space and practically no supervision. It can also be employed for existing arc suppression coils which are adjustable under load. The design and method of operation of this gear will doubtless make it very popular in electricity supply circles and may largely prove instrumental in inducing those operating engineers, who with a certain amount of justification have so far declined to protect their systems by arc suppression coils, to adopt this method of earth-fault protection.

(MS 954) *A. van Gastel. (E. G. W.)*

THE BROWN BOVERI PUMPLESS AIR-COOLED MUTATOR.

Decimal Index 621.314.65.032.11

This article briefly mentions the salient aspects and operating features of the pumpless mutator, together with details of operating experience to date.

THE introduction of a new or modified product on the market is only warranted when advantages accrue to the purchaser. These may comprise enhanced efficiency, greater reliability, simplification of plant, decreased maintenance, longer life, or reduced initial outlay.

In certain cases one or other or a combination of the foregoing advantages may even justify a higher purchase price.

A new product which combines a number of advantages with reduced cost is the pumpless mutator. Like its predecessor with pumps it must also be able continuously to cope with varying operating conditions, i. e., sustain short circuits and, under certain conditions, also backfires without detriment or impairment of the vacuum necessary for reliable operation. To meet this requirement the mutator must be carefully designed, the tank and electrode entries rendered permanently vacuum-tight, and the interior of the walls made capable of absorbing gases. The latter property is necessary, inasmuch as, notwithstanding suitable pre-treatment of the mutator components, slight quantities of gas may be liberated from the electrodes and arc guide tubes in the event of heavy overloads, short circuits, or backfires occurring, with possibly unpleasant consequences in the course of time.

After exhaustive tests pumpless mutators satisfying these conditions have now been put on the market. These are of the air-cooled type (Figs. 1 and 2) with two different possible methods of cooling. Fig. 3 shows the mutator without the outer jacket for conducting

Figs. 1—3. — Pumpless air-cooled mutators type L 06.

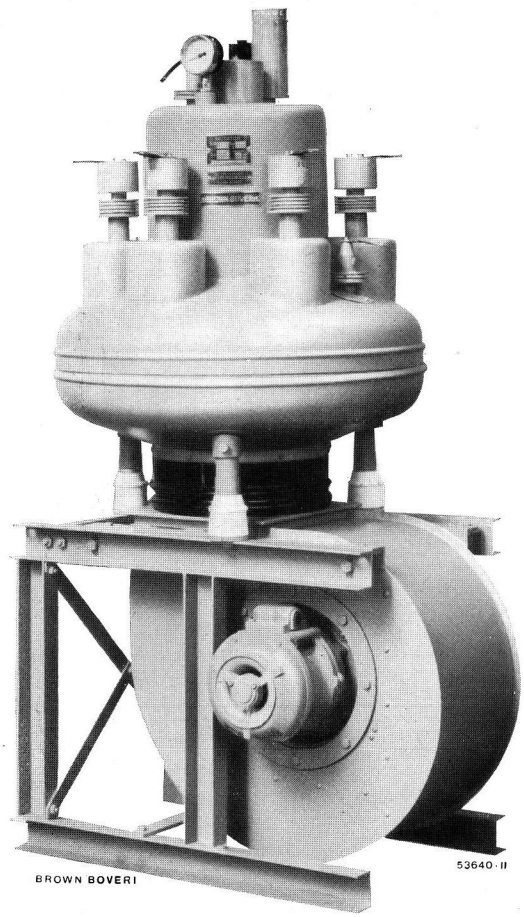


Fig. 1. — Standard design with centrifugal fan. Pumps and vacuum equipment are dispensed with, thus rendering this ideal small convertor extremely simple.

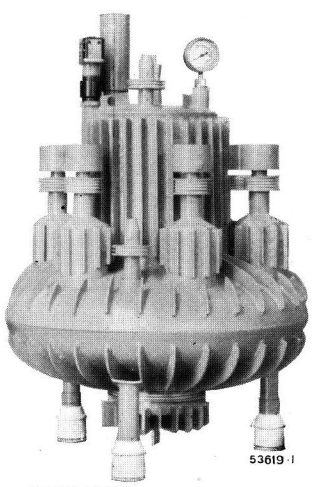


Fig. 3. — As Figs. 1 and 2, but without jackets for air cooling. The cooling surface is increased by means of ribs on the mutator casing.



Fig. 2. — Unit with axial-flow fan as built during development stage.

the air. Its extremely simple and robust design is particularly evident from this illustration. No porcelain is employed in the electrode entries. The iron parts of the electrodes are insulated from one another by a glass-like mass, which also forms a vacuum-tight joint. This material has the same coefficient of expansion as the iron, so that even should the electrode entries become over-heated no harmful stresses will be set up and cause the fused mass to crack. The electrode entry seals and insulation are in any case so far removed from, and so well protected against the arc, that it is impossible for them to become excessively hot in service. In order to accelerate dissipation of the heat generated in the mutator the cooling surface has been substantially increased by means of welded-on ribs. The electrodes are made of graphite which is thoroughly de-gassed and purified by annealing under vacuum at high temperature before being fitted in the mutator.

For the so-termed "formation" process a pump set is coupled to the mutator and the fan allowed to run. The mutator is connected as under normal working conditions and loaded with the rated current, over-currents, and short circuits until the electrodes and all parts in contact with the arc cease to generate a measurable quantity of gas. Thereafter, the mutator is heated in a stove at several hundred degrees Centigrade until its walls also cease to give up a quantity of gas which can be measured. When cold, or more correctly, at the working temperature, the walls of mutators treated in this manner are again capable of absorbing considerable quantities of gas. [1]

The mutator is then so well formed that it can be used continuously without pumps for any working conditions. The vacuum pipe, which hitherto connected the mutator to the air pumps, is squeezed together immediately above the mutator tank, welded up, and cut off just above the weld. The unit is thereupon ready for service.

A number of such mutators was subjected to various load tests at their designed voltages, both immediately after being welded up and again after lengthy storage. Such tests, however, cannot compare with continuous tests under practical operating conditions. For this

[1] Cf. also Dr. Deck: "The Adsorption of Gases by the Steel Walls of Vacuum Discharge Apparatus and its Bearing on the Life of Pumpless Mutators". The Brown Boveri Review 1942, page 202.

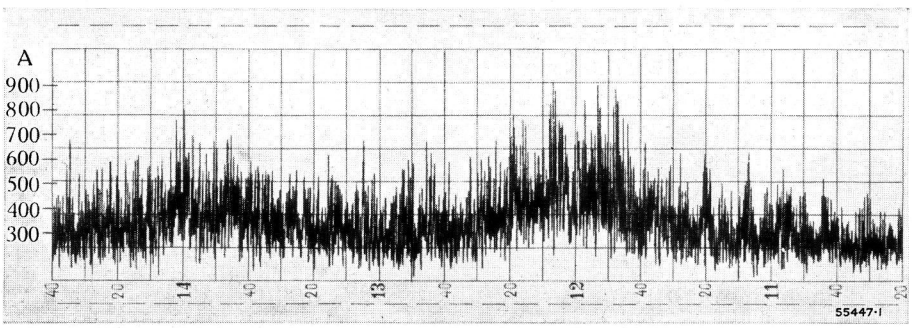


Fig. 4. — Portion of load diagram of Basle Electricity Supply Voltastrasse Substation.
D. C. voltage 600 V. A fairly high mean load is followed by high overload peaks of up to 900 A or more.

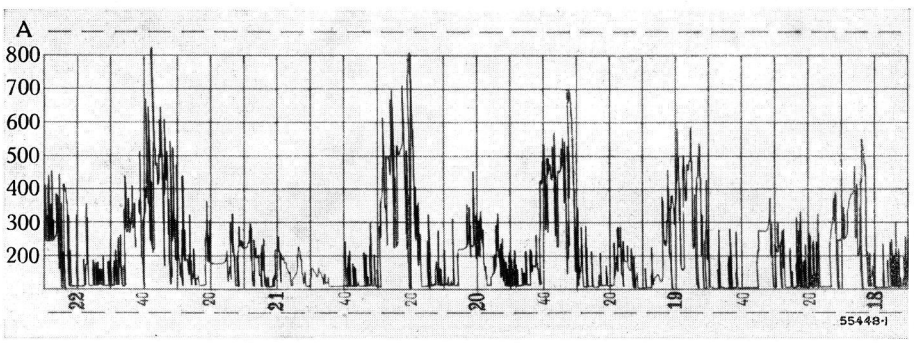


Fig. 5. — Portion of load diagram of Bremgarten Substation on the Dietikon-Bremgarten-Wohlen Railway.
D. C. voltage 850 — 950 V. The characteristic feature of local railway operation is the low basic load with high load peaks.

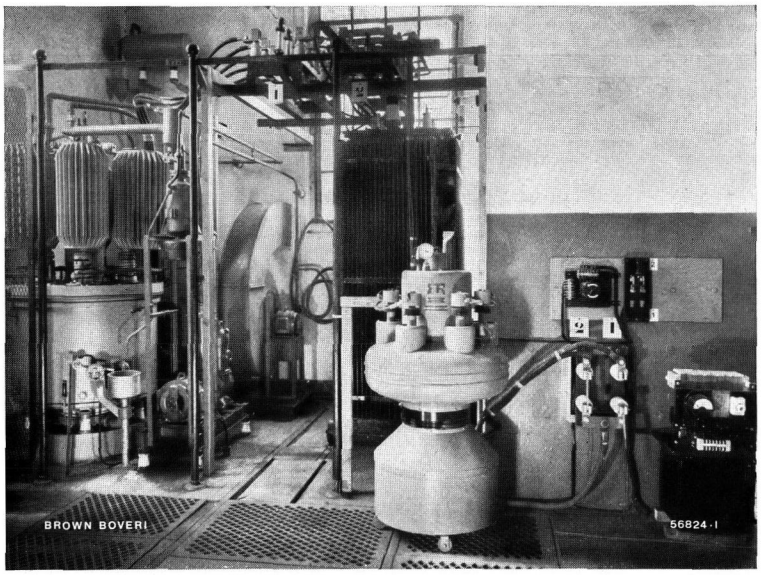


Fig. 6. — Pumpless air-cooled mutator type L 06.
Trial unit in the Bremgarten Substation on the Dietikon-Bremgarten-Wohlen Railway. Comparison with the adjacent existing station mutator of the same rating shows the small space requirements of the new set.

reason, pumpless mutators were installed in substations where they could take over the service of existing pumped mutators and where load conditions were as varied as possible.

Figs. 4 and 5 show portions of load diagrams of two such substations, the first representing typical, intensive tramway service at 600 V d. c. and the second the load of a small local railway with a d. c. voltage of 850—950 V. Fig. 6 shows the trial mutator provisionally installed in a substation on the local railway.

The first trial units were put into service at the beginning of 1940. A number of small imperfections, such as are invariably inherent in most new equipment, gradually became evident, but these fortunately proved to have nothing to do with the vacuum and were easily remedied in the units built subsequently. Since then no further trouble has been experienced with the trial mutators during periods of up to two and a half years' continuous service.

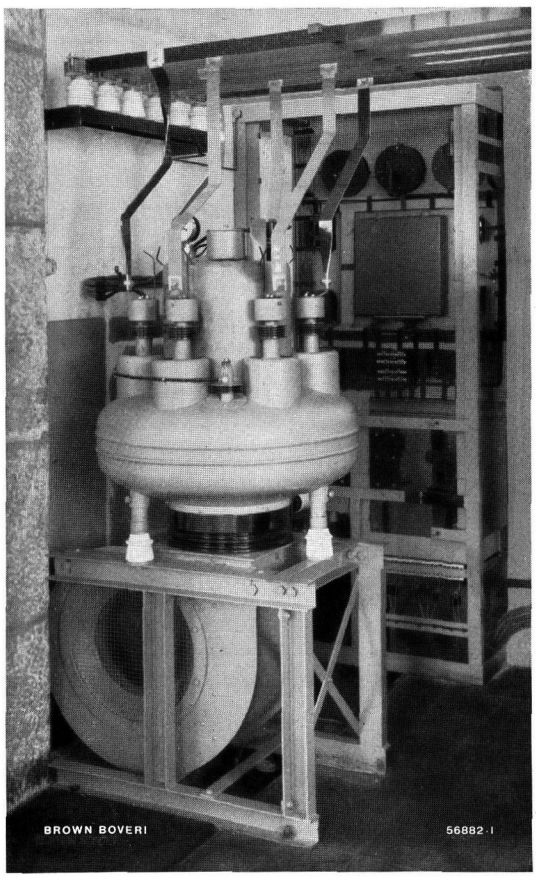


Fig. 7. — Pumpless air-cooled mutator type L 06.
Installed in the Pontebrolla Substation of the Società Ferrovia Regionale Ticinese, Locarno. Rating 400 kW, 1350 V, 296 A.

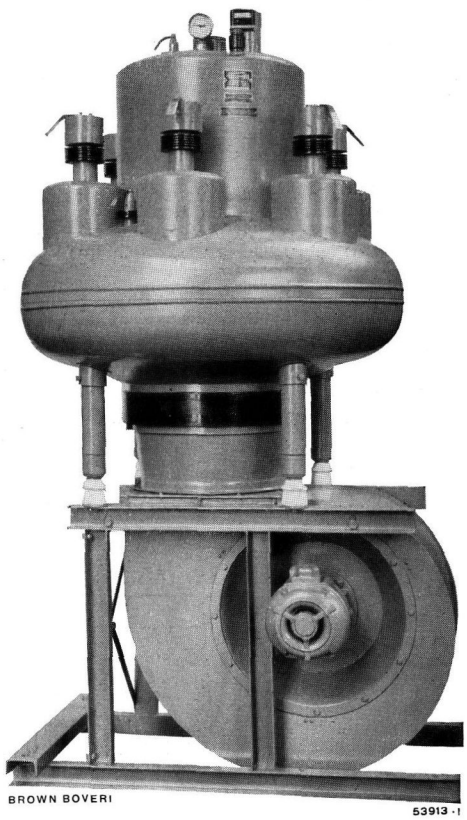


Fig. 8. — Pumpless air-cooled mutator type Lg 36.
Rating 900 kW, 1500 A at 600 V. Combines extreme simplicity of design and operation with small space requirements and high efficiency.

In view of the relatively short service periods of 1—2 1/2 years it is impossible to give a reliable estimate of the life of a pumpless mutator, but from the gas adsorption tests already referred to a very long service period can be reckoned with.

Apart from the fact that the rotary and high-vacuum pumps, and the vacuum pipe, vacuum cock, and vacuum meter are dispensed with pumpless air-cooled mutators have the following noteworthy advantages:— High efficiency, low price and weight, small space requirements, simplicity of layout, no cooling water, no maintenance, no supervision, no re-forming prior to setting to work, always ready for service, and great reliability.

The first pumpless mutator to be developed was the type L 06 rated 300 kW, 500 A at 600 V. The overload rating conforms to class A of the I. E. C., i.e., 25 % for 15 minutes, 50 % for 2 minutes, 100 % for 10 seconds.

A large number of this type is already in service, both in Switzerland and abroad. Fig. 7 is a view of a plant installed in a confined space. All installations have so far been put into service without a hitch and operate trouble-free.

In the interim the development of the next larger pumpless air-cooled mutator (Fig. 8) has been completed. This has controlled grids and bears the type designation Lg 36. It is rated 900 kW, 1500 A at 600 V and is designed to sustain overloads as per class A of the I. E. C. A mutator of this type has also been in trial operation in a tramway system for some time past.

To date over thirty pumpless mutators either have been supplied or are being built to order. The development of pumpless mutators represents a further advance and brings about a long-needed simplification of the plant.

(MS 951) *S. Widmer. (E. G. W.)*

BRIEF BUT INTERESTING

New Battery-operated Mine Locomotives.

Decimal Index 621.335.3—835

ACCORDING to service conditions, petrol, crude-oil, steam, and compressed-air locomotives, as well as battery-operated vehicles are to be found on industrial, contractors', and mine railways.

Many years' experience, however, has shown that in the majority of cases battery-operated vehicles are superior to the other forms of traction, both from an economic and technical point of view.

Their economy is a result of the low energy costs — the battery being charged during the night hours or at other times when excess energy is available at low rates — little maintenance required, and cleanliness of operation.

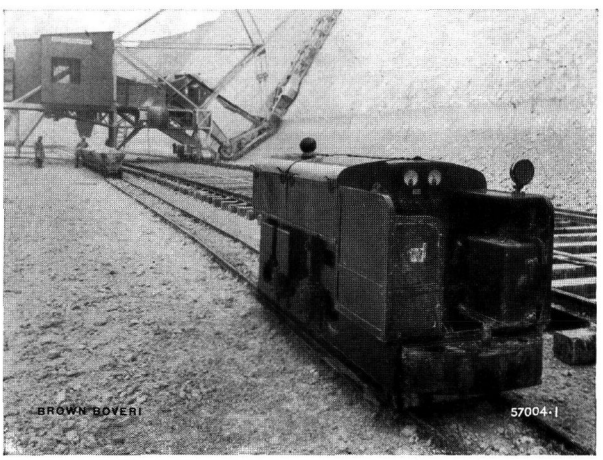


Fig. 1. — 5 H. P. mine locomotive for 500 mm gauge track.

Radius of action about 37 km on the level with a trailing load of 8 t and an eighty-cell battery with a capacity of 111 Ah for three hours.

Electric mine locomotives are economic and reliable.

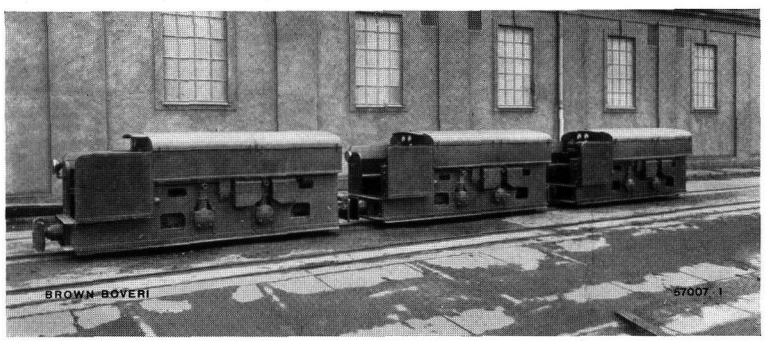


Fig. 2. — 10 H. P. mine locomotive for 500 mm gauge track.

Radius of action about 37 km on the level with a trailing load of 17 t and an eighty-cell battery with a capacity of 165 Ah for three hours.

Compact, robust construction and modern electrical equipments characterize these locomotives.

From the technical point of view the superiority of the battery-operated vehicle accrues from the simplicity of its construction, together with its ease of operation, constant availability, and extreme reliability. It is true that the initial outlay is often greater than with other forms of traction, but the extra expense involved

can be written off within a short time, due to the reduced working expenses.

These long-known advantages of battery-operated vehicles have led to their being widely adopted for industrial, contractors', and mine railways. In numerous cases, however, battery-operated locomotives have also been employed, and this type of vehicle doubtless still has a big future before it.

Battery-operated mine locomotives with modern Brown Boveri equipments are illustrated in Figs. 1 and 2. These locomotives, which were put into service for the transport of coal and chalk in a cement works at the beginning of 1942, have proved so satisfactory that several further locomotives have already been ordered by the same customer.

(MS 924) *G. Manta. (E. G. W.)*

Scandinavian Locomotives with Brown Boveri High-voltage Control Gear.

Decimal Index 621.337.1.027.3

VIRTUALLY all high-power locomotives now built in *Switzerland* are fitted with Brown Boveri high-voltage control gear, which permits big savings to be realized in switchgear and cables, as well as important raw materials to be economized, due to the fact that the currents to be controlled are much lower than on the low-voltage side. Moreover, the high-voltage control gear can be more readily accommodated in the locomotive, since notwithstanding the much greater number of regulating steps, which enable the locomotive to be smoothly started and its speed to be progressively regulated without complicated designs having to be resorted to, it is much less bulky. Another advantage is that the on-load tap-changing switch and the transformer are so intimately combined that the switch contacts operate under oil in the transformer tank and only need inspection on the occasion of the general overhaul of the locomotive, i. e., every three or four years.

These features, together with the proved reliability of the Brown Boveri high-voltage control gear, are meeting with greater and greater appreciation abroad. Fifteen high-voltage-controlled locomotives are already in service or under construction in *Norway*, while the management of the *Swedish State Railways*, which like the Swiss Federal and Norwegian State Railways are operated at 15,000 V, $16^2/_3$ cycles single-phase, has also decided to equip a new series of express locomotives with high-voltage control by reason of the simplification, reduced overall weight, lower maintenance costs, and smooth regulation accruing

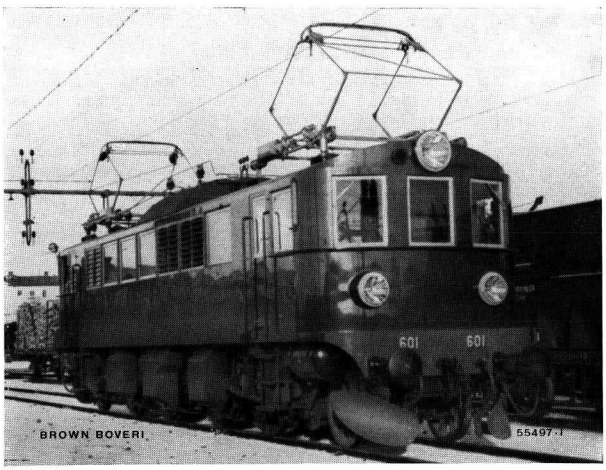


Fig. 1. — Swedish State Railways trial locomotive type 1 Do 1 No. F 601 with Brown Boveri high-voltage control gear.

With this control gear fine regulation is obtained for starting and speed variation, while big savings in weight and space requirements are realized. The maintenance costs are reduced due to the on-load tap-changing switch being combined with the transformer and operating under oil. Contact wear is restricted to two arcing switches not immersed in the oil and which only have to rupture relatively low currents.

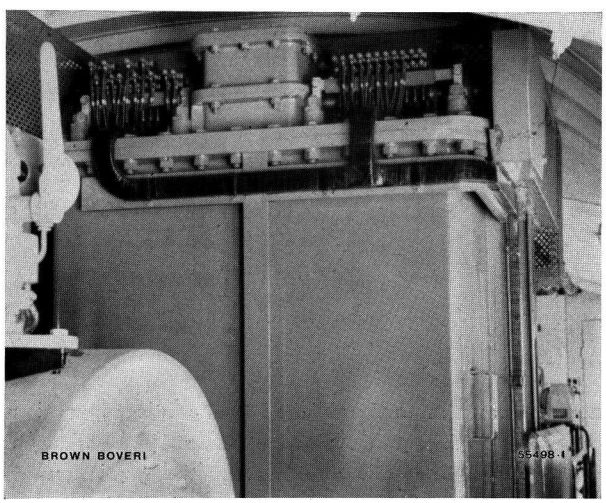


Fig. 2. — Locomotive transformer with Brown Boveri high-voltage control gear on the Swedish State Railways trial locomotive F 601.

The control gear and its motor drive form an integral part of the transformer.

from the special transformer with twenty-eight regulating steps supplied from our Baden works and fitted by the Swedish State Railways on their trial locomotive No. F 601 in 1942.

(MS 995) *F. Gubler. (E. G. W.)*

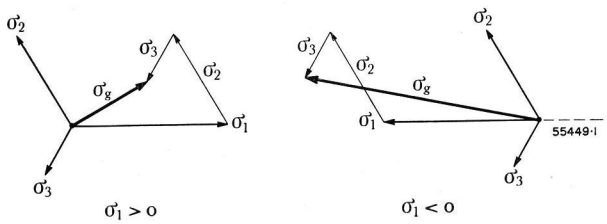
A Contribution to the Application of the Theory of constant Energy of Deformation.

Decimal Index 539.4.001.2

IN order to determine the strength of a design from the theory of constant energy of deformation the so-termed "comparative stress" must be determined from the formula

$$\sigma_g = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1 \sigma_2 - \sigma_2 \sigma_3 - \sigma_3 \sigma_1}$$

where σ_1, σ_2, and σ_3 are the principal stresses.



The three principal stress components are plotted 120° apart and added vectorially. Negative values are drawn from the origin in the opposite direction.

σ_1, σ_2, σ_3, Principal stresses. σ_g, "Comparative stress".

From this formula a diagram can be constructed with σ_1, σ_2, and σ_3 as vectors 120° apart. The resultant of *vectorial* addition is then the "comparative stress" σ_g.

(MS 991) *A. Meldahl. (E. G. W.)*

Squirrel-cage Motor takes lowest Starting Current!

Decimal Index 621.313.332.2
621.316.717

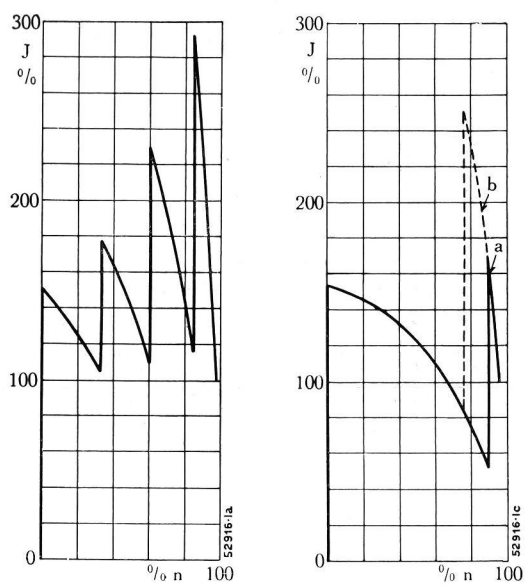
FOR years past the connection of squirrel-cage motors to public supply systems has been hampered by severe restrictions, and when the enormous efforts made by the electrical industry in every country during this long period, the numerous treatises published, the countless inventions patented, and the untold devices put forward in an endeavour to find a motor which would satisfy the regulations on the score of starting current, are recalled, one is tempted to wonder whether the policy followed was really the correct one.

Over 90 % of the motors connected to public supply systems drive machines such as centrifugal pumps, fans, machine-tools, agricultural machines, etc., which are started up light, i. e., the load is not thrown on until the motor has run up to speed. In such cases star-delta starting is particularly indicated, inasmuch as the initial current rush is only 150—200 % of the rated current, a value to which no power supply undertaking can possibly raise any objection.

Although no claim is made for the origination of this method of starting, a reliable switch was hitherto lacking which could not be prematurely changed over from star to delta, i. e., before the motor has time to attain its full revolutions. This requirement is met with the ingenious type OS star-delta motor protecting switches marketed by Brown Boveri, which permit the starting time to be adjusted between 3 and 240 seconds and are available for both hand and solenoid operation. Once the switch is closed the star-delta starting process is carried out entirely automatically and correctly without the intervention of the attendant, so that at the moment of change-over the current surge is limited to the lowest value theoretically possible (see curves).

The following example affords striking proof of the superiority of such control gear:—

A Brown Boveri 12.5 kW squirrel-cage motor with star-delta motor protecting switch was substituted for a 9 kW motor with centrifugal starter which had been supplied by another firm and proved to be too small for the beating mill to which it had been allocated. Whereas the original motor caused current peaks of 60—65 A at starting, those involved by the approximately 40 % larger motor were



Comparison of starting currents.

On left: Motor with centrifugal starter.

On right: Squirrel-cage motor with correctly designed star-delta starter,

a. Starting against 40 . . . 50 % torque.

b. Fan or pump drive.

Squirrel-cage motors started with type OS star-delta motor protecting switches, which **automatically** and **correctly change over connections** at the most favourable moment, cause much less pronounced starting peaks than motors with centrifugal starter. **Even for large outputs** it is therefore more advantageous to employ **the simple, inexpensive squirrel-cage type of motor** equipped with our type OS starting switch instead of a motor with wound armature, as long as the torque developed during the star-delta starting process suffices for the drive in question.

much less pronounced, i. e., only of the order of 50 A. What is more, the larger motor, including the star-delta starter, is even less expensive than the original equipment. This case shows that it is to the mutual interest of both the electricity supply and the consumer to employ, wherever possible, squirrel-cage motors with a star-delta switch designed to control the starting process correctly.

In industrial plants conditions are fundamentally different, inasmuch as large distribution transformers prevent fluctuations on the consumer's side inordinately impairing the stability of the electricity supply system, so that even large squirrel-cage motors can generally be thrown directly across the line.

(MS 998) *S. Hopferwieser. (E. G. W.)*

tween Mörel (Goms Valley) and Turtmann power stations providing the h. f. transmission channel. A repeater installed at the latter station is connected to a P. B. X. equipped with approximately 250 lines at the Chippis works. In this manner, rapid communication between all departments of the undertaking is always possible.

This carrier-current telephone system, which can also be equipped for remote metering purposes, gives entire satisfaction and greatly facilitates the operation and supervision of the extensive power station and distribution equipment.

The accompanying illustrations show the two-wave h. f. aluminium-wound choke coils rated 600 A, which are connected to the high-voltage line to block the passage

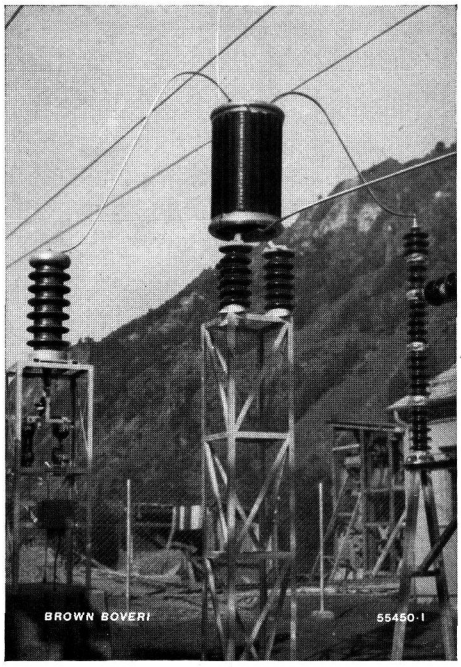


Fig. 1. — Mörel Power Station.

In centre, h. f. choke coil mounted on framework. On right, framework of lightning arrestors. On left, 75 kV coupling condenser with protective apparatus and line-matching unit.

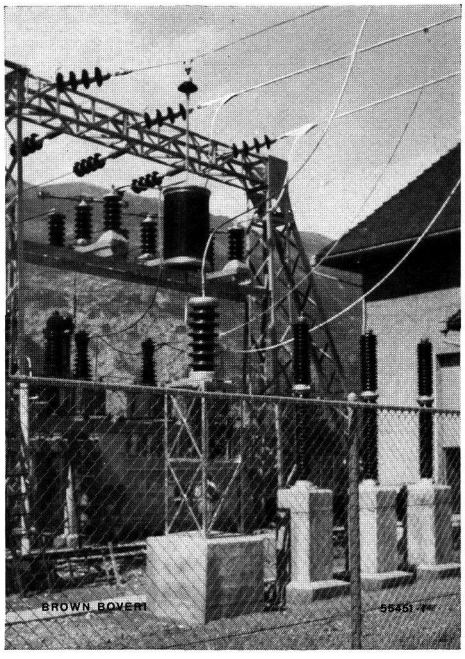


Fig. 2. — Turtmann Power Station.

In centre, h.f. choke coil suspended from coupling phase. Underneath in foreground coupling condenser with protective apparatus and line-matching unit on framework.

The aluminium windings of these two-wave h. f. choke coils are designed for a relatively high service current of 600 A and a short-circuit current of 7000 A.

Setting to Work of a Carrier-current Telephone Installation.

Decimal Index 621.396.44

A carrier-current telephone installation ordered by the Rhonewerke A.-G. was successfully put into operation at the beginning of August, the 65 kV power line between Mörel (Goms Valley) and Turtmann power stations providing the h. f. transmission channel. A repeater installed at the latter station is connected to a P. B. X. equipped with approximately 250 lines at the Chippis works. In this manner, rapid communication between all departments of the undertaking is always possible.

This carrier-current telephone system, which can also be equipped for remote metering purposes, gives entire satisfaction and greatly facilitates the operation and supervision of the extensive power station and distribution equipment.

The accompanying illustrations show the two-wave h. f. aluminium-wound choke coils rated 600 A, which are connected to the high-voltage line to block the passage of two carrier waves. The one at Mörel is mounted on a special framework, while the other at Turtmann is suspended directly from the coupling phase of the high-voltage system. Alongside the choke coils, the 4350 cm condensers connecting the h. f. equipment to the power line are also visible.

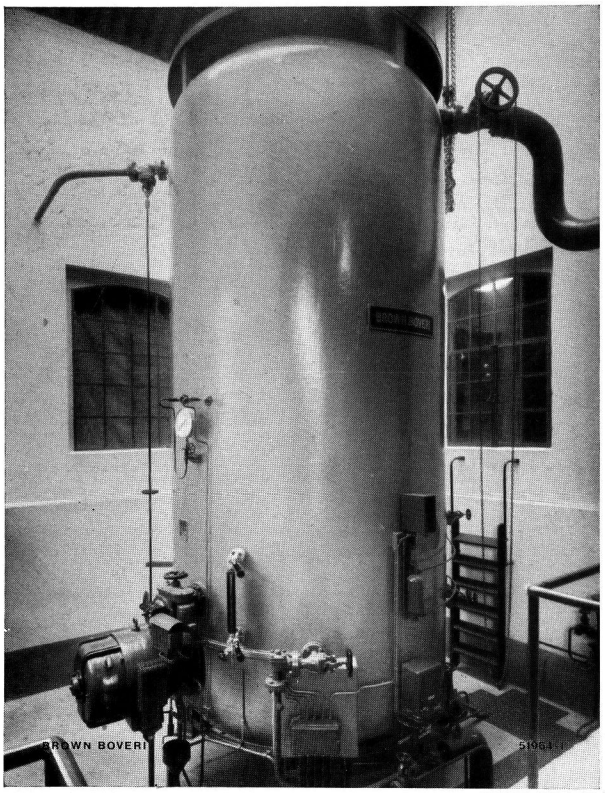
(MS 990) *E. Hancess. (E. G. W.)*

Reliability of Brown Boveri Electric Boilers.

Decimal Index 621.181.646

FOR many years past Brown Boveri have built high-voltage boilers on the water-jet principle, i. e., water-jets convey the current from one electrode to the other. Progressive load variation is possible from no-load to full-load through regulation of the jets, this involving no moving parts inside the boiler. Ceramic insulating materials are kept out of the boiler water where they would be subject to attack from alkalis. In consequence, the boiler is very reliable and requires little maintenance. Proof of this statement is furnished by a 6000 kW water-jet electric boiler connected to a single-phase 15,000 V supply system, which was overhauled after a period of three years' service. The electrode system showed no traces of wear, while the boiler itself was still quite clean. As a result it was possible to re-assemble it and put it into service again without delay.

(MS 985) *A. Spoerli. (E. G. W.)*



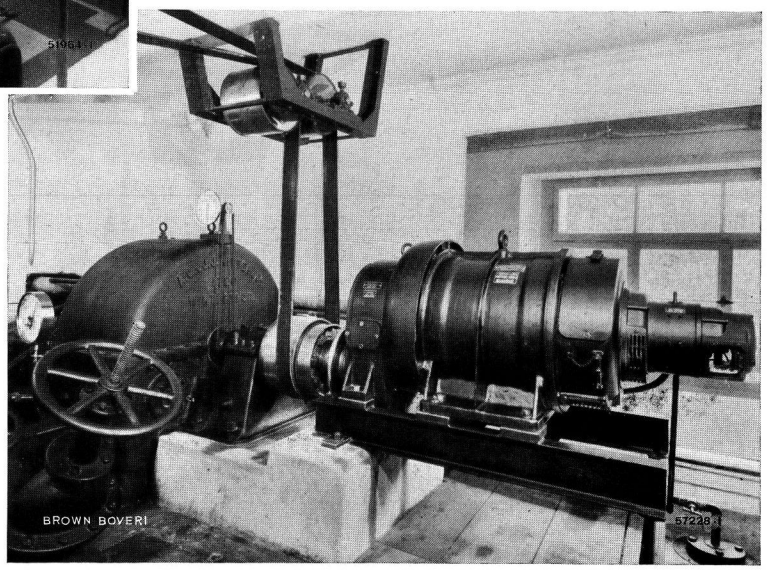
6000 kW water-jet electric boiler for 15,000 V single-phase. Progressive regulation without moving parts and the absence of ceramic insulators in the boiler water render this boiler extremely reliable.

Synchronous Machine saves more than 100 t of Coal annually.

Decimal Index 621.34 : 664.7
621.313.32

ABOUT three years ago, owing to the difficulties experienced in obtaining fuel, Messrs. Heussi & Michel, Mühlehorn (Switzerland), substituted a three-phase synchronous machine for a steam engine which up till then had been used to supplement the hydraulic power of their flour mill. The machine is readily controlled, requires very little attendance, and during periods of reduced water flow can immediately be put into service for driving purposes. It saves the mill 100—110 t of coal annually. In summer and autumn, to exploit excess hydraulic power, the machine is run as generator and supplies electric motors, cookers, and radiators, as well as a private dehydration plant which has been made available to the inhabitants of Mühlehorn.

(MS 931) *E. Baertschi. (E. G. W.)*



Drive of flour mill owned by Messrs. Heussi & Michel, Mühlehorn (Switzerland) comprising a high-pressure water turbine of about 125 kW and a 90 kW, 1500 r. p. m. three-phase synchronous machine connected to the turbine shaft through built-on reduction gearing and clutch. At the other end the turbine shaft is coupled to the mill lineshafting.

Sales of Air-blast High-speed Circuit-breakers.

Decimal Index 621.316.57.064.45

THE Brown Boveri indoor-type air-blast high-speed circuit-breaker was put on the market in 1935, and the outdoor type three years later. Towards the end of 1942 the order for the four-thousandth air-blast circuit-breaker

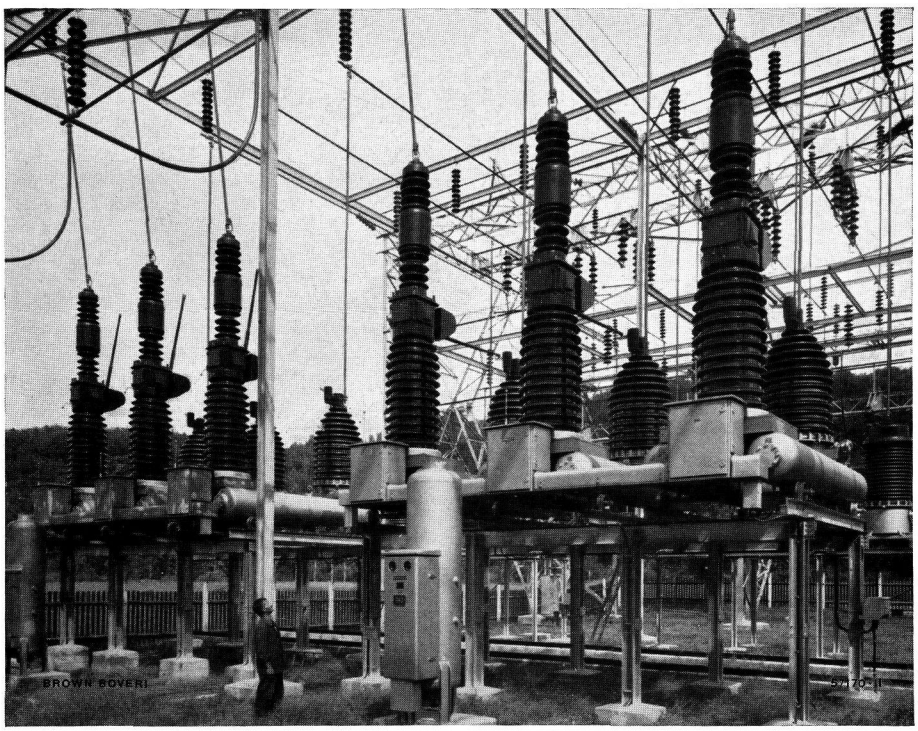
was booked. Of these 4000 circuit-breakers about 3500 were of the indoor, and the remaining 500 of the outdoor type. These figures are proof of the favourable reception met with by this new, entirely unconventional type of circuit-breaker. Its design and the resulting operating advantages, such as elimination of fire and smoke hazard for the whole plant, extremely short rupturing time, slight wear and easy replacement of contacts, freedom from creepage paths, etc., are indeed attractive. An additional advantage of the indoor circuit-breaker is its adaptability to any layout, inasmuch as it can be mounted in practically any desired position similar to an isolating switch. This permits the design of readily supervised, simple installations requiring no complicated cubicles and oil drains.

On the other hand, a circuit-breaker depending on the availability of compressed air, both for its operation and the extinction of the arc, could at first not but arouse certain misgivings amongst operating engineers, for at the time of its introduction they had had little experience with compressed-air generating and distributing installations. In consequence, it was doubted whether the latter would fulfil the standards of reliability required of electrical installations. Experience soon showed, however, that, if correctly laid out, compressed-air plants were just as reliable as the circuit-breakers themselves and required very little attendance.

Among the 3500 circuit-breakers for indoor erection there are types for all voltages from 6 to 87 kV, rupturing capacities from 100 to 1500 MVA, and rated currents from 400 to 5000 A. The extremely numerous repeat orders from customers having had Brown Boveri air-blast high-speed circuit-breakers in operation for more or less lengthy periods prove that the users have recognized the outstanding advantages of this type of switchgear.

The outdoor circuit-breakers supplied include types for voltages between 20 and 220 kV and are erected in plants with extremely varied operating and climatic conditions (see accompanying illustration). Tentative designs intended for erection out of doors were tried out for a period of several months under winter conditions, artificially rendered more severe, thousands of feet above sea level, on the heights of the Jungfraujoch and the Säntis. It was found that the inherent advantages of the indoor type were also highly desirable for outdoor circuit-breakers. In addition, the possibility of high-speed reclosing, particularly in the case of high and extra high voltage systems proved of great interest. A large number of outdoor air-blast circuit-breakers has since been provided with the high-speed reclosing feature from the beginning, but the device can be fitted subsequently, if necessary. It is to be anticipated that Brown Boveri air-blast high-speed circuit-breakers will play a bigger and bigger part in network protection.

(MS 932) *L. Burckhardt. (E. G. W.)*



150 kV outdoor-type air-blast high-speed circuit-breaker in a Swiss installation.

Like the circuit-breakers themselves the method of mounting has gone through a long period of development. At present, elevated erection is generally preferred, since it precludes the risk of accidental contact and permits free movement about the station, as well as ready supervision. Additional air storage tanks, if required, are installed as close as possible to the circuit-breakers and can be combined with the housing containing the valves and pressure gauges.



Published by BROWN, BOVERI & COMPANY, LIMITED, BADEN (Switzerland). Printed by Kreis & Co., Basle, Switzerland.
Obtainable through A. Francke A. G., Berne and Rouge & Cie., Lausanne (Switzerland).