

3,040,528

Filed March 21, 1960



BY

Karl F. Ross
AGENT.

1

3,040,528

VAPOR TURBINES

Harry Zvi Tabor, Beth Hakerem, Jerusalem, and Lucien Bronicki, Katamon, Jerusalem, Israel, assignors to the State of Israel

Filed Mar. 21, 1960, Ser. No. 16,256

Claims priority, application Israel Mar. 22, 1959

1 Claim. (Cl. 60—36)

The present invention relates to vapor turbines based on the Rankine cycle.

Up to now it has been the general practice to operate vapor turbines by means of steam. The main advantage of steam is that its temperature-entropy diagram is such that the slope of the branch marking the vapor/liquid boundary is negative, i.e.

$$\frac{dS}{dT} < 0$$

(The boundary in question will herein be referred to as vapor/liquid boundary although in fact it is a boundary between the state in which the liquid and the vapor co-exist and a state in which there exists only vapor.) This means that when saturated steam expands isentropically the effluent is within the region in which steam and liquid water co-exist. In other words, the effluent steam is not superheated and the theoretical efficiency of a Rankine cycle, carried out with steam, is accordingly satisfactory. This, however, entails the result that in a steam turbine, particularly a small single-stage turbine, the steam, saturated at the beginning, is wet after expansion in the nozzle. If this wetness exceeds a certain amount it can cause erosion of the turbine blades, which is a serious disadvantage.

A further disadvantage of the use of steam resides in the fact that even for moderate enthalpy (total heat) drops the efflux velocity from the nozzles is very high, making it very difficult or impossible to operate a single-wheel turbine at the correct speed which, as known, has to be about half the efflux velocity. In addition it is known that for small turbines, i.e. for low-H.P. turbines, the blades and the nozzle must be extremely small, which is another source of considerable inefficiency since experience shows that the efficiency of a turbine stage decreases rapidly as the size of the blades is reduced.

The efflux velocity of a vapor (for expansion between two fixed temperatures) is to a first approximation inversely proportional to the square root of the molecular weight, or, in other words, the higher the molecular weight of the fluid the smaller the efflux velocity. Hereinafter fluids of higher molecular weight will be referred to as "heavy."

Because of the above approximate relation between the efflux velocity and the molecular weight, it has frequently been suggested to use heavy vapors for the operation of turbines. The use of such heavy vapors would indeed remove the above-mentioned disadvantages inherent in steam. Thus, for example, a vapor nine times as heavy as steam, i.e. of molecular weight of 162, will give efflux velocities of the order of one-third of that of steam so that a single turbine stage will suffice in many cases or, in the case of a large number of stages, the number may be reduced to one-ninth (pressure staging) or to a third (velocity compounding) for the same speed and diameter of the rotor. Another advantage of the use of heavy fluids results from the fact that for the same output a larger mass of vapor must pass through the turbine as a result of which the size of the blades and with it the efficiency of the turbine is increased.

From the above it becomes apparent that for small turbines where one wishes to avoid the complication of

2

multiple stages, or even velocity compounding, and at the same time to increase the size of the nozzles and blades for better efficiency the use of a heavy vapor is advantageous as compared with steam. Furthermore, if the boiling point of the vapor is high, i.e. the density of the vapor in which the blades rotate is lower than for steam, the disc and bucket friction losses for a given speed can be reduced below those for steam.

However, in the use of heavy fluids for the operation of a turbine an inherent difficulty is encountered resulting from the fact that the slope of the liquid/vapor boundary of the temperature-entropy diagram of fluids usually changes from negative to positive as the number of atoms in the molecule is increased, i.e.

$$\frac{dS}{dT}$$

becomes greater than zero. The implication of this will now be explained with reference to the drawing in which: FIGS. 1 and 2 are temperature-entropy diagrams of water and of a heavy fluid, respectively; FIG. 3 is a diagrammatic view of an apparatus for carrying out the process of the invention.

When carrying out a Rankine cycle with steam starting from point 1, the water is first heated from T_1 to T_2 (FIG. 1) with the entropy being increased accordingly. (For the sake of simplicity the compression of the liquid is here ignored.) At T_2 , which corresponds to point 2 on the curve, steam is generated isothermally; that part of the cycle is represented by the stretch 2, 3. The points 2, 3 lie on the branches of the diagram marking the water/water-steam-mixture boundary and the water-steam-mixture/steam boundary, respectively. From point 3 the next stage of the cycle consists in an isentropic expansion which in an ideal case proceeds along the stretch 3, 4. However, in practice the expansion is never quite isentropic and, allowing for frictional losses, a state is reached which is represented by point 4'.

In FIG. 2, 1 is again the starting point in the Rankine cycle, the starting temperature being T_1 . The liquid is heated to T_2 and its state at this temperature is represented by point 2. The stretch 2, 3 represents vaporization. In the case of ideal expansion, the expansion is represented by a line 3, 4 normal to the abscissa; in practice, however, ideal conditions are not realized in a turbine and the path is along line 3, 4'. This means that by carrying out a Rankine cycle with a heavy fluid, expansion yields superheated vapor resulting in low theoretical efficiency of the cycle and the necessity for the condensers to desuperheat before condensing.

It is accordingly the object of the present invention to devise a method for overcoming the above disadvantage inherent in heavy fluids, and thereby to adapt the latter for the operation of turbines based on the Rankine cycle.

In some cases, however, it also happens that a superheated effluent is obtained with fluids of which the slope of the temperature-entropy diagram on the liquid/vapor boundary is zero or even negative. This can be the result of special circumstances such as, for example, high friction inside the turbine.

Therefore, in a more general way it is the object of the present invention to provide a method for the operation of turbines based upon the Rankine cycle in which the vapors leaving the exhaust are normally superheated.

The invention consists in a method for the operation of a turbine based on the Rankine cycle and in which the exhaust vapor is in a superheated state, wherein the exhaust vapor is made to give off heat to the feed fluid.

An aspect of the invention consists in a turbine adapted to be operated by the above method comprising near the exhaust a heat exchanger adapted to effect the heat transfer from the exhaust vapor to the feed fluid.

Finally, a further aspect of the invention consists in a power-generating unit comprising a vapor turbine based upon the Rankine cycle and operable with saturated vapors of such fluids and/or under such conditions which yield superheated exhaust vapors, characterized in that a heat exchanger is provided for transferring the superheat from the exhaust vapors to the feed liquid.

According to a preferred embodiment of the invention, the desuperheating heat exchanger is of the counterflow type. Because of bad heat transfer from superheated vapors, the heat exchanger has, moreover, to be of as large a contact surface as possible and/or reconcilable with other considerations.

Because the heat exchange takes place between the liquid and gaseous phases of one and the same fluid, the vapors can only be cooled down to saturation in case of an ideal heat exchanger while in practice the vapor leaving the desuperheating heat exchanger will still be slightly superheated. For this reason it is in some cases advantageous to insert a cooler between the heat exchanger and the condenser, if any, in order to economize on the latter.

As pointed out before it may happen owing to special circumstances that the exhaust vapors are superheated in the case of fluids in which

$$\frac{dS}{dT}$$

of the liquid/vapor boundary of the temperature-entropy diagram is zero or even negative, while when a fluid is used whose

$$\frac{dS}{dT}$$

on the liquid/vapor boundary is positive the exhaust vapor is invariably in a superheated state. The invention is applicable in both cases.

Moreover, the invention is applicable to both condensing and non-condensing turbines, although in most cases, in particular when fluids other than water are used, condensing turbines are preferred.

The invention is illustrated, by way of example only, in the accompanying FIG. 2 referred to above and FIG. 3 which is a diagram of a turbine power unit in accordance with the present invention.

From FIG. 2 it can also be seen that, as a result of the exchange between superheated vapors and the feed liquid, the former is desuperheated down to a point 5 which lies on the temperature-entropy diagram. This desuperheating proceeds theoretically along the constant-pressure line P_1 which is either the pressure inside the condenser in case of a condensing turbine or atmospheric pressure in case of a non-condensing turbine. However, in a real heat exchanger in which there exists a small pressure drop on the vapor side the desuperheating proceeds along a slightly different path P_1' shown as 4''5'5 where 5' is the exit condition from the desuperheating heat exchanger and the cooling represented by the stretch 5'5 takes place in the inlet zone of the condenser, if any, or in a cooler inserted for the purpose. (Points 5 and 5' approach one another as the temperature differential at the cold end of the heat exchanger, i.e. between the cooled vapors and the liquid to be heated, is reduced. Possible pressure differences of the vapor at the inlet and at the outlet of the heat exchanger, due to changes in area of flow, have been ignored for reasons of simplicity. Likewise, compression of the feed liquid has been ignored.) During the desuperheating of the vapor the quantity of the heat given off is equal to the area 4'5 C D (in the case of an ideal heat exchanger). In such an ideal heat exchanger this entire heat is given off to the feed liquid so that the latter is heated from T_1 to T_3 , T_3 being defined by the point B on the left-hand side of the diagram and the latter being determined by the fact that the areas A B F E and 4'5 C D (transferred heat) have to be equal to each other.

The benefit derived from such a heat exchanger is very large. In fluids such as octane operating over a range of

150° C. to 40° C., the amount of energy fed back by this heat exchanger may be of the order of 30% of the total energy required by the boiler so that the cycle efficiency will be of the order of 30% more than if the heat exchanger were omitted.

This method is not to be confused with the regenerative feed heating as used in a steam turbine. In such a system steam is bled off the turbine at various points during the expansion and used to preheat the feed. In this manner the steam bled off does not continue through the turbine so that it does not complete its work possibilities while the throttle steam flow is increased, thereby increasing the feed-pump work. Contrary thereto the vapors used in accordance with the invention are exhaust vapors, i.e. they are used at a stage at which they already have performed their work while they are at the same time in a superheated state, i.e. in addition to their unrecoverable latent heat of condensation they contain heat which can be recuperated.

A further advantage of the invention is that it allows the frictional heat produced in the turbine to be saved and fed to the boiler in contrast to the steam cycle where the friction heat is "wasted" in vaporizing some of the wetness in the final steam mixture which subsequently has to be condensed anyhow. The system of the invention is particularly advantageous in a single-stage turbine where normally there is no reheat factor and where, in accordance with the invention, the emerging friction heat may be a considerable part of the total heat supply. Even the final kinetic energy of the exhaust—which is usually called exhaust loss—can be caught in the heat exchanger and usefully employed to preheat the feed.

It is found in many cases that the efficiency of such a cycle using a fluid giving a superheated exhaust and using a good heat exchanger is larger than with a fluid whose vapor after passage through the turbine is wet so that no desuperheating heat exchanger can be used.

FIG. 3 illustrates diagrammatically a unit according to the invention which in this case is a condensing unit. It comprises a turbine 10 coupled with an electric power generator 11, a boiler 12, a counterflow heat exchanger 13, a cooler 14, a condenser 15 and a pump 16. The liquid is vaporized in the boiler and the vapors pass into the turbine which is thereby operated. After leaving the exhaust the superheated vapors pass through the heat exchanger 13 where they give off their superheat to the recycled feed liquid. The de-superheated vapors then enter the cooler 14 and from there pass into the condenser 15 where they are liquefied. The resulting liquid is pumped by means of pump 16 through the heat exchanger 13 back into the boiler 12. In many cases the cooler 14 may be omitted and the cooling of the desuperheated vapors is effected in the first part of the condenser 15. Both cooler 14 and condenser 15 are cooled by cooling water or air in any conventional manner.

The choice of the working fluids is quite wide and depends upon the operating temperatures and the molecular weight deemed necessary. Thus the higher paraffins such as octane (normal or iso) are found to be very satisfactory as they appear to be very stable in contact with the usual materials of construction (iron, aluminum, copper, etc.). The heavier aromatics and ethers are also quite suitable. For very small turbines, where the disc and windage losses can be serious and the nozzles and blades are very small, a higher-boiling liquid is chosen (such as monochlorobenzene or dichlorobenzene) so as to reduce the losses and increase the nozzle and blade size. For larger turbines, where the disc and windage losses are, in many cases, small in proportion to the output, somewhat lower-boiling fluids are chosen (such as iso-octane), in order to avoid having to make nozzles and blades unnecessarily large.

We claim:

Method of generating power, comprising the steps of:

5

- (a) circulating in a closed circuit a fluid having a molecular weight greater than that of water and a boiling point at atmospheric pressure in excess of substantially 90° C., said fluid being selected from the group which consists of octane, paraffins having a molecular weight in excess of that of octane, and chlorinated benzenes while having a temperature-entropy diagram wherein the slope of the line designating the liquid-vapor/vapor boundary is positive; 5
- (b) vaporizing said fluid at saturation temperature; 10
- (c) expanding in a turbine the saturated vapor so produced, thereby superheating said vapor;
- (d) removing enough sensible heat from the resulting superheated effluent emanating from the turbine to de-superheat said effluent; 15
- (e) condensing the de-superheated effluent to produce a liquid;

6

- (f) preheating said liquid with the heat removed from said effluent; and
- (g) repeating the cycle by again vaporizing the preheated liquid at saturation temperature.

References Cited in the file of this patent

UNITED STATES PATENTS

1,688,413	Delas	Oct. 23, 1928
2,303,159	Cross et al.	Nov. 24, 1942
2,471,476	Benning et al.	May 31, 1949
2,707,239	Riehl	Apr. 26, 1955
2,793,502	Riehl	May 28, 1957

FOREIGN PATENTS

280,920	Germany	Dec. 3, 1914
---------	---------	--------------