

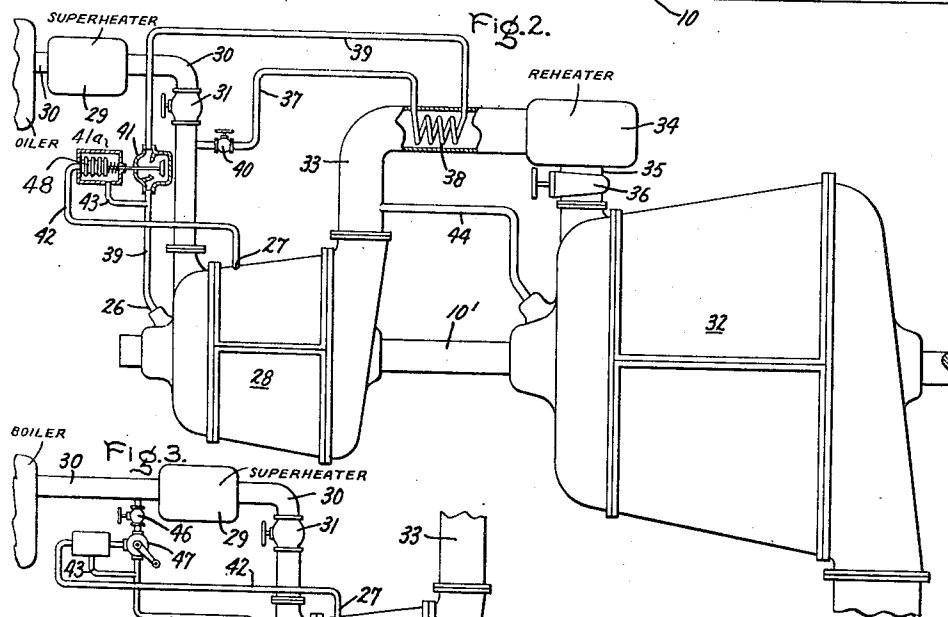
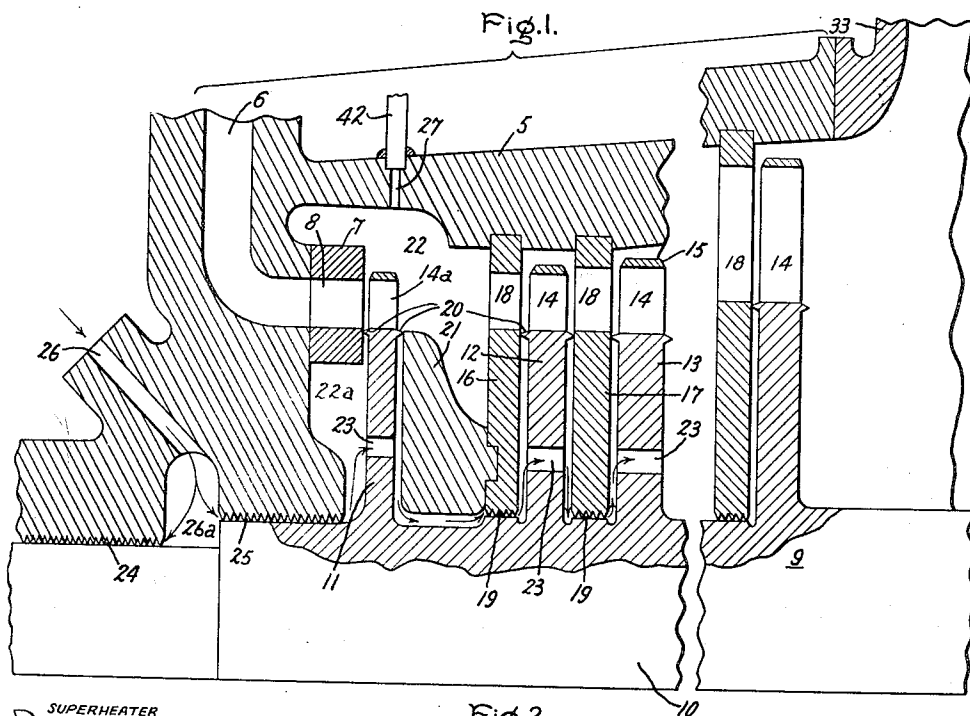
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TURBINE ROTOR COOLING ARRANGEMENT

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TURBINE ROTOR COOLING ARRANGEMENT

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This invention relates to a method of cooling high temperature elastic fluid turbines, especially the initial stages of a multi-stage turbine. More particularly, it relates to an arrangement for circulating a cooling fluid through a high temperature elastic fluid turbine in order to cool the bucket wheels and shaft of the turbine. The invention constitutes a further improvement of the turbine sealing and cooling arrangement disclosed by U. S. Patent 1,878,731, issued September 20, 1932 in the name of Paul W. Thompson.

This is a division of an application, Serial Number 706,310, filed October 29, 1946, now Patent No. 2,451,261, in the names of Glenn B. Warren and Arthur R. Smith and assigned to the same assignee as the present application.

As is well known, high thermal efficiency can be attained in a fluid pressure turbine prime mover by utilizing a high temperature, high pressure fluid operating medium, such as highly superheated steam at temperatures over 1000° F. Special materials, such as the so-called "18-8" wrought chrome-nickel steel, which can safely withstand these high temperatures are commercially available, but their cost is such that it is economically impractical to use these materials throughout a turbine, especially in large multi-stage axial flow turbines, where the shaft and bucket-wheels are ordinarily machined from a single forging. The allowable working stress for ordinary, less expensive materials, such as "chrome-moly-vanadium" steels when subjected to temperatures of the order of 1000° F. is considerably below the stresses encountered in present day turbines. However, it is found that by decreasing the normal working temperature of chrome-moly-vanadium steel, for instance from 1100° F. to 900° F., the allowable creep stress may be increased by about five times its former value. Since the permissible working stress increases rapidly as a function of decrease in temperature, readily available low cost materials can be used for the turbine rotor, provided the rotor is maintained at a temperature such that the allowable stress is greater than the maximum working stresses encountered. The more expensive special materials can then be used for the buckets or blades in the first few stages where the temperature conditions are most severe. To obtain effective cooling action with a reasonable quantity of cooling fluid, the temperature of the coolant should preferably be at least 150° below that of the motive fluid, when the latter is supplied to the turbine at temperatures over 800° F. The invention is intended to provide a cooling arrangement which will permit the use of ordinary materials in turbines using a high temperature fluid operating medium.

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Accordingly, an object of the invention is to provide an improved method of and arrangement for cooling high temperature turbine parts, particularly the rotor.

5 Another object is to provide a turbine arrangement which permits increasing the normal operating temperatures of turbines manufactured from ordinary low cost materials.

10 A further object is to provide means for keeping the stresses in a steam turbine rotor within allowable safe limits while operating at temperatures of the order of 1000° F. and above.

Other objects and advantages will be apparent from the following description taken in connection with the accompanying drawing, in which
15 Fig. 1 is a partial sectional view of an elastic fluid turbine of a type which may advantageously employ the invention; and Figs. 2 and 3 are diagrammatic views of elastic fluid turbine power-
20 plants embodying the invention.

For convenience in explanation, the invention shall be described as applied to a steam turbine powerplant; however, it should be understood that the invention may be also applicable to other
25 elastic fluid turbines, such as those using mercury vapor, products of combustion, or other high temperature operating media.

Referring now to Fig. 1, the turbine comprises a casing 5 defining a steam inlet conduit 6 communicating with a first-stage nozzle ring 7 having a plurality of circumferentially spaced nozzles 8 and secured by suitable fastening means (not shown) to casing 5. A rotor indicated generally at 9 is disposed within casing 5, supported by
30 suitable bearings (not shown), and comprises a shaft 10 and a plurality of bucket wheels, 11, 12, 13, which may be formed integral with shaft 10 from a single forging. Secured to the circumferences of the bucket wheels are a plurality of spaced buckets or blades 14 surrounded by shroud
35 bands 15. Stationary diaphragms 16, 17 are supported in casing 5 and associated with wheels 12, 13 for directing operating medium to the respective bucket annuli. These diaphragms are in the form of annular disc members containing a plu-
40 rality of circumferentially spaced blades 18 forming nozzle passages. Arranged around the central openings in the diaphragms and cooperating with the shaft 10 are labyrinth packings 19 which limit the flow of steam through the clearance
45 spaces between the shaft 10 and the respective diaphragms. Fig. 1 is intended to represent diagrammatically a general type of diaphragm arrangement, many specific forms of which are well
50 known in the steam turbine art.

55 Circumferential flanges or "spill bands" 20 are formed on either side of the first stage wheel 11 adjacent the root of buckets 14a, and are

arranged to form close axial clearances with nozzle ring 7 on the inlet side and a ring member 21 on the discharge side, the latter being supported by the diaphragm 18. Ring 21 also forms the inner wall of the steam distribution chamber 22 between the first stage wheel 11 and the second stage diaphragm 16. Similar spill bands on subsequent stage wheels 12 and 13 are arranged to cooperate with portions of the respective adjacent diaphragms 16, 17, etc. The purpose of the spill bands will be noted hereinafter.

In order to prevent excessive axial forces on rotor 9, pressure balancing holes 23 are provided through the web portions of the respective wheels, in a manner well known in the art. Formed in casing 5 are axially spaced labyrinth seal packing ring structures 24 and 25, which form close clearances with shaft 10 for resisting the flow of steam axially along the shaft. Provided in casing 5 is an admission port 26 for sealing and cooling fluid, communicating with an annular chamber 26a defined in the casing between the labyrinth seals 24, 25. Seal 25 is located between chamber 26a and a chamber 22a defined between casing 5 and the first stage wheel 11 and within nozzle ring 7.

For reasons which will appear hereinafter, a pressure tap port 27 communicating with chamber 22 may be provided in casing 5.

The operation of the turbine of Fig. 1 is as follows. Superheated steam enters the turbine casing from a valve chest (not shown), thence through passage 6 to the first stage nozzles 8 and axially through the respective buckets 14a, 14 and diaphragm nozzles 18, imparting rotational energy to the rotor 9. In so-called "impulse stages" of this general arrangement, substantial pressure drops are experienced by the operating medium in passing through the first stage nozzles 8 and the subsequent diaphragm nozzles 18. Some drop in pressure may occur in the buckets of the several wheels, but the major portions of the pressure drop across any given stage occurs in the stationary nozzle for that stage.

In order to cool the turbine, cooling and sealing steam at a temperature considerably lower, but at a pressure substantially equal to or somewhat greater than that of the operating steam in main inlet 6, is admitted through port 26 to the annular chamber 26a. From chamber 26a, some of this cool steam will flow axially outwardly between the shaft and seal 24, thus helping to cool shaft 10. This portion of the cooling steam which flows outwardly may be collected in any one of several well-known ways and led to some lower pressure stage of the turbine or to a suitable heat reclaiming device (not shown). The above-described arrangement is more particularly described and claimed in United States patent to P. W. Thompson 1,878,731.

Because of the material drop in pressure across nozzles 8, chambers 22 and 22a will both be at pressures appreciably below that in chamber 26a; therefore some cooling steam will flow axially inwardly between the shaft and labyrinth seal 25. This part of the cooling steam which flows inwardly past seal 25 will fill the chamber 22a adjacent the first stage wheel 11. The spill bands 20 resist the mixing of the hot operating fluid from nozzles 8 with the cooling steam in space 22a, therefore the cooling steam, following the path of least resistance, will flow through the respective balance holes 23, and between the labyrinth seals 19 and shaft 10 through subse-

quent stages in the manner indicated by the arrows in Fig. 1. This circulation of cooling steam through the turbine effectively cools the shaft, bucket wheels, and the central portions of the diaphragms in the high temperature section of the turbine.

The spill bands 20 do not entirely prevent the mixing of the hot and cool fluids, some cooling steam leaking past the spill bands into the flow path of the operating steam at each stage. It is important, from the standpoint of both thermal and aerodynamic efficiency, that provision be made to keep this leakage in the first few stages to a minimum. To accomplish this, the spill band clearances in these stages are made very small, so that most of the entry of cooling steam into the motive fluid occurs in the lower pressure stages, where the temperature of the operating medium is more nearly equal to that of the cooling steam. Furthermore, the static pressure of cooling fluid in chamber 22a is roughly of the same order of magnitude as that of the motive fluid at the first stage bucket inlet (and in chamber 22) because of the pressure drop in packing 25. Therefore, there is no marked tendency for the cooling fluid to enter the first stage bucket flow path past sealing rings 20.

The cooling steam may be obtained in various ways, one of which is illustrated in Fig. 2. This represents a so-called "tandem compound turbine" powerplant comprising a high pressure steam turbine 28 and a low pressure turbine 32, which may have rotors connected to a common shaft 10'. Each of the turbines 28, 32 are arranged internally as shown in Fig. 1. The high pressure turbine 28 is supplied with superheated steam, which is generated in a boiler and then passes through supply conduit 30 containing a superheater 29 and a suitable shut-off valve 31, thence to the turbine. From the high pressure turbine 28, exhaust steam passes through the "cross-over conduit" 33 to a reheater 34, where the temperature is raised somewhat, thence through conduit 35 and stop-valve 36 to the inlet of the low pressure turbine 32.

Cooling fluid for the high pressure turbine 28 is obtained by taking superheated steam from conduit 30 at a point downstream from the shut-off valve 31 (i. e. between valve 31 and the inlet to turbine 28), and passing it through a conduit 37 to a suitable cooling coil 38 contained within the cross-over conduit 33. The action of the comparatively lower temperature exhaust steam from turbine 28, passing over the exterior surfaces of the cooling coil 38, reduces the temperature of the superheated steam within the coil considerably, while the pressure remains substantially constant. From coil 38, the cooled steam passes through conduit 39 to the cooling fluid inlet port 26, thence through the turbine 28 in the manner described above.

It is preferred that the cooling steam be taken from conduit 30 at a point downstream from the shut-off valve 31, since with this arrangement, if a turbine overspeed condition should occur, the action of the emergency overspeed governor (not shown) will cause shut-off valve 31 to close automatically, thereby stopping the supply of both the operating steam and the cooling steam.

The flow of cooling steam to coil 38 may be controlled by a manually controlled valve 40 in conduit 37. The flow of the cooled steam to the turbine 28 may be further controlled by a valve 41 in conduit 39. While valve 41 may be manually controlled, it is preferred that a suitable

automatic valve be used. For instance, a type of valve well-known as a "constant pressure differential valve" may be advantageously employed. This is arranged to maintain constant the pressure differential between supply conduit 39 and the chamber 22 of Fig. 1. To this end a conduit 42 connects pressure tap 27 in the turbine casing with a pressure responsive member such as flexible bellows 48 contained in a housing 41a, while another conduit 43 connects housing 41a with conduit 39 at the discharge side of valve 41. As may be seen in Fig. 2, the stem and disk of valve 41 is attached to bellows 48 so that a pressure decrease in chamber 22 of Fig. 1 will cause bellows 48 to contract thereby moving the valve disk towards the closed position with the result that the fluid pressure in conduit 39 between valve 41 and port 26 decreases proportionately. Conversely a pressure increase in chamber 22 is accompanied by a proportionate pressure increase in conduit 39.

Cooling steam for the low pressure turbine 32 is advantageously obtained by extracting low temperature exhaust steam from cross-over conduit 33, before it enters the reheater 34, and passing it through conduit 44 to the annular chamber 26a of the turbine 32.

It should be noted that, ordinarily, means for controlling the rate of flow and pressure in the conduit 44 will be unnecessary, since the exhaust steam pressure in cross-over conduit 33 varies automatically with changes in the load on turbine 32. The pressure differential between the inlet port 26 and the chamber 22a of turbine 32 will be substantially equal to the sum of the pressure drops across the reheater 34, valve 36, nozzles 8, and the friction losses in the related conduits. As the load increases the rate of flow of operating steam increases, with a proportionate increase in this pressure differential. This increase in the pressure differential automatically causes the rate of flow of cooling steam through conduit 44 to increase, thus providing the increased cooling effect needed for the higher load condition. With this arrangement, the pressure tap 27 of Fig. 1 is obviously not needed and would be omitted or suitably blanked off.

Another method of obtaining the cooling steam for the high pressure turbine 28 is shown in Fig. 3. Here again turbine 28 is supplied with high temperature steam from superheater 29 through conduit 30 containing a suitable shut-off valve 31, as in Fig. 1. Substantially saturated steam, considerably lower in temperature and slightly higher in pressure than the superheated operating steam supplied to the turbine, is taken from conduit 30 at the inlet side of superheater 29 and passes through conduit 45 to the cooling steam inlet port 26, thence through the turbine 28 in the manner described above. Control of the cooling steam is here effected by a suitable shut-off valve 46 and a constant pressure differential valve arrangement 47 in conduit 45. The constant pressure differential valve 47 may be arranged as described above, so as to maintain a constant differential between the cooling fluid inlet port 26 and the chamber 22 surrounding the first stage wheel 11 of turbine 28.

The arrangement illustrated in Fig. 2 is particularly desirable in that the cooling steam can be taken from the supply conduit 30 at a point comparatively near the turbine, whereas the arrangement of Fig. 3 requires a pipe line 45 running all the way back to the boiler (superheater 29 ordinarily being incorporated in the boiler

which supplies the operating steam), which in large modern powerplants may be several hundred feet from the turbine.

It will be seen that the invention provides an improved arrangement for effectively cooling high temperature steam turbines, permitting the use of ordinary low-cost material for operation with considerably increased normal operating temperatures, while stresses in the rotor are maintained within safe limits.

What I claim as new and desire to secure by Letters Patent of the United States, is:

1. In a high pressure multi-stage elastic fluid turbine for high temperature operation, the combination of a casing having a first motive fluid inlet port and a second cooling fluid inlet port, a rotor disposed in the casing and having a shaft with a plurality of axially spaced wheels each with a circumferential row of buckets and at least one axial pressure balancing passage through the wheel at a location radially inward from the buckets, a plurality of nozzle diaphragms supported in the casing between the respective bucket-wheels and each having packing means surrounding the shaft and defining a restricted fluid flow path from one side of the diaphragm to the other, said wheels and diaphragms having circumferential portions located between the buckets and the axial pressure balancing passages and forming seals defining restricted flow paths communicating between the motive fluid flow path and the spaces defined between the respective diaphragms and wheels radially inward of said fluid seals, the casing having walls defining a passage for conducting motive fluid from said first inlet port to the first stage wheel and means defining a second passage from the second inlet port to the inlet side of the first stage wheel at a location radially inward from the circumferential fluid seal, said axial pressure balancing passages and the restricted flow paths through the diaphragm packing means forming parts of a cooling fluid flow path communicating with said second passage.

2. A turbine in accordance with claim 1 in which the effective area of the flow paths through said circumferential fluid seals is as small as practicable in the initial stages of the turbine whereby the mixing of the cooling fluid with the motive fluid is kept to a minimum, the effective area of the flow paths through said fluid seals in subsequent stages being larger, whereby the cooling fluid is caused to enter the motive fluid path in said subsequent stages.

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REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
1,421,087	Johnson	June 27, 1922
2,213,940	Jendrassik	Sept. 3, 1940
2,243,467	Jendrassik	May 27, 1941
2,487,514	Boestad et al.	Nov. 8, 1949
2,489,683	Stalker	Nov. 29, 1949

FOREIGN PATENTS

Number	Country	Date
195,736	Switzerland	July 16, 1938
196,931	Great Britain	Jan. 17, 1924