

Dec. 10, 1957

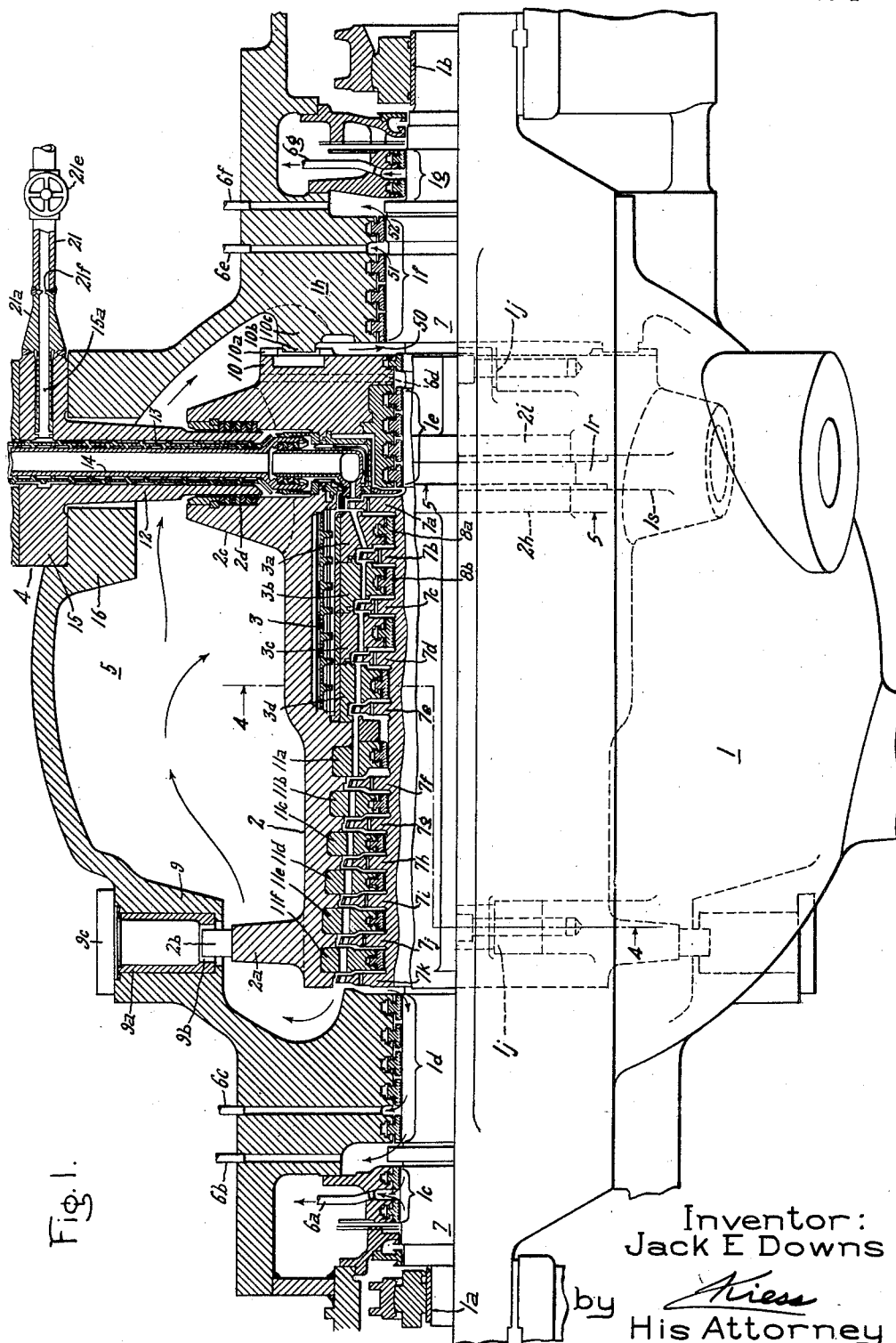
J. E. DOWNS

2,815,645

SUPER-CRITICAL PRESSURE ELASTIC FLUID TURBINE

Filed March 1, 1955

4 Sheets-Sheet 1



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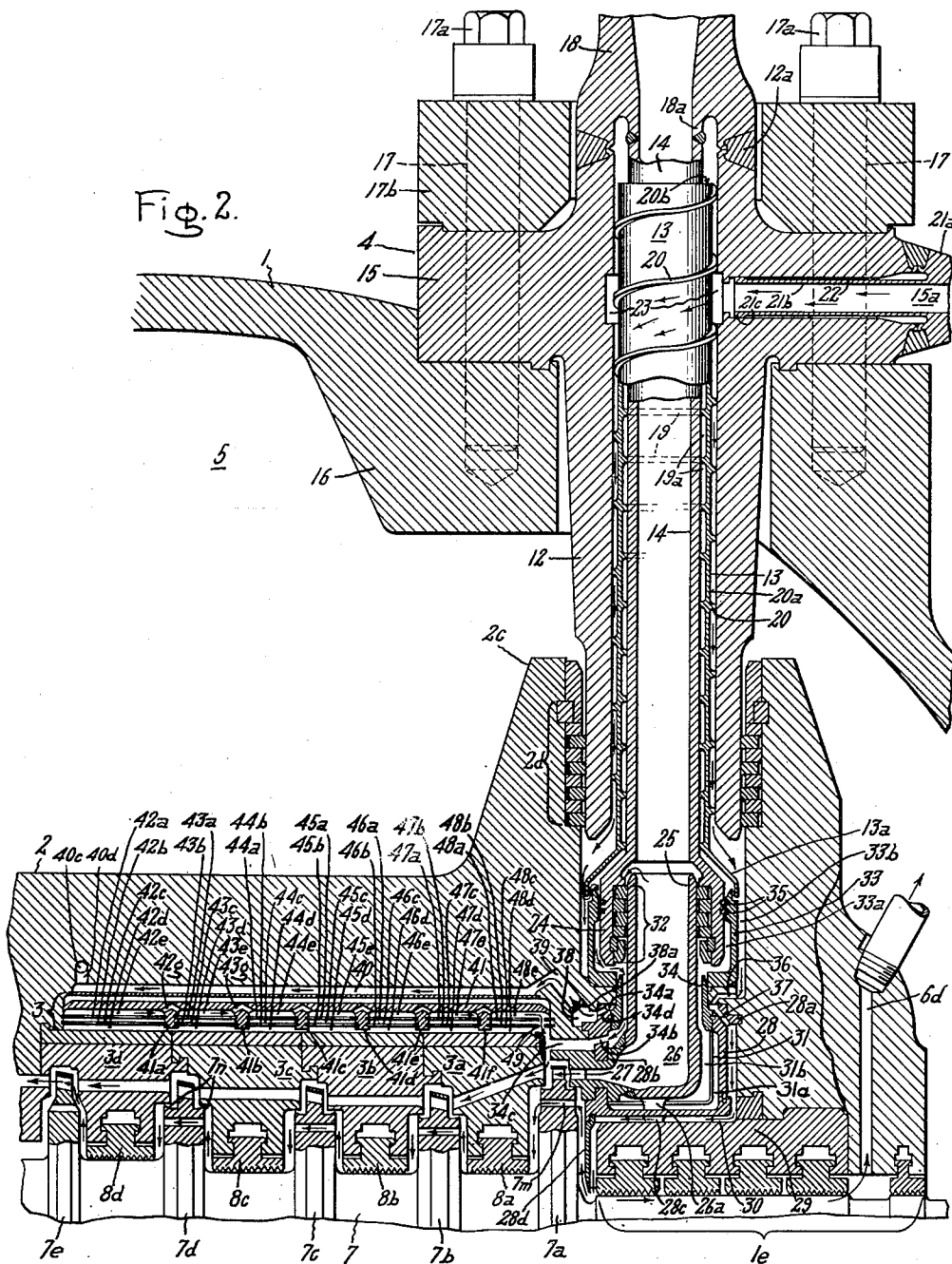
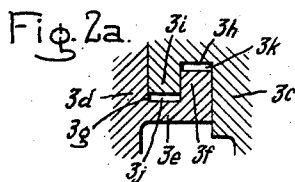
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SUPER-CRITICAL PRESSURE ELASTIC FLUID TURBINE

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4 Sheets-Sheet 2



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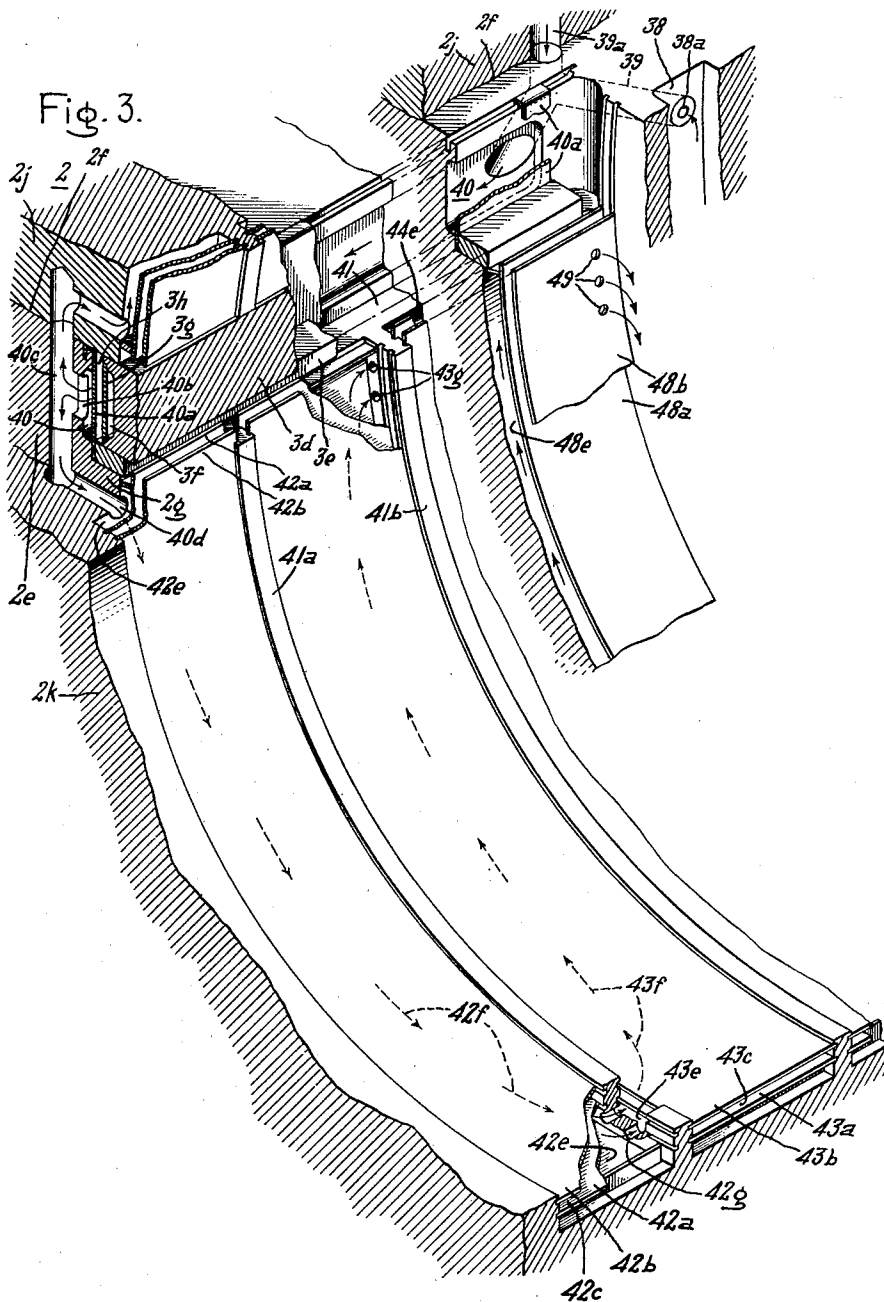
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SUPER-CRITICAL PRESSURE ELASTIC FLUID TURBINE

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4 Sheets-Sheet 3



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SUPER-CRITICAL PRESSURE ELASTIC FLUID TURBINE

Filed March 1, 1955

4 Sheets-Sheet 4

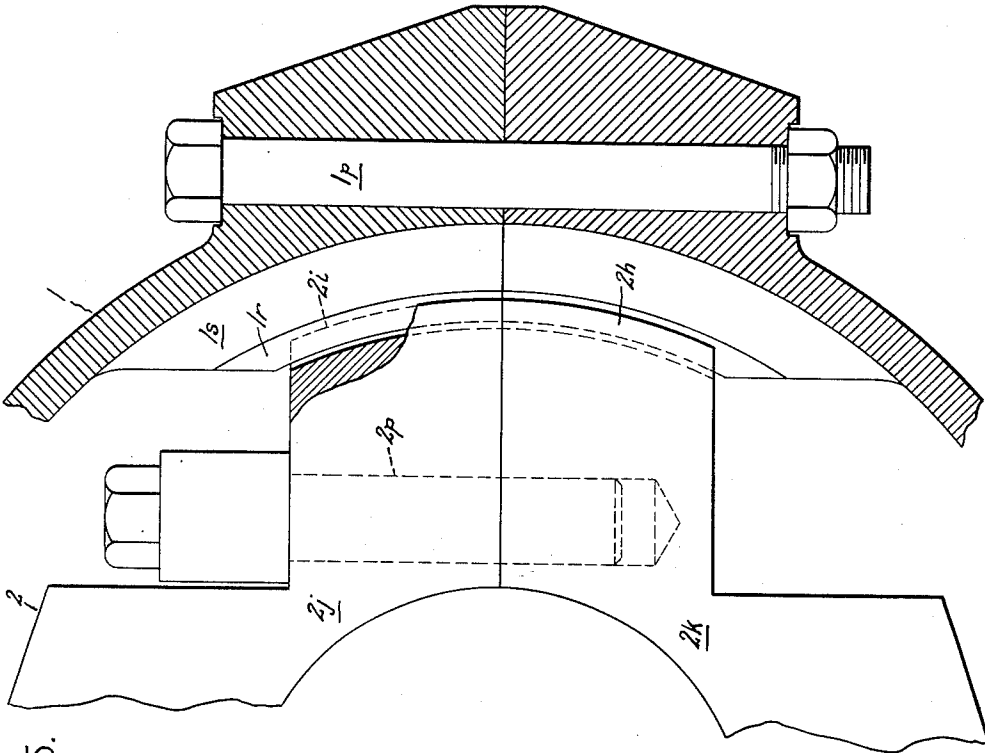


Fig. 5.

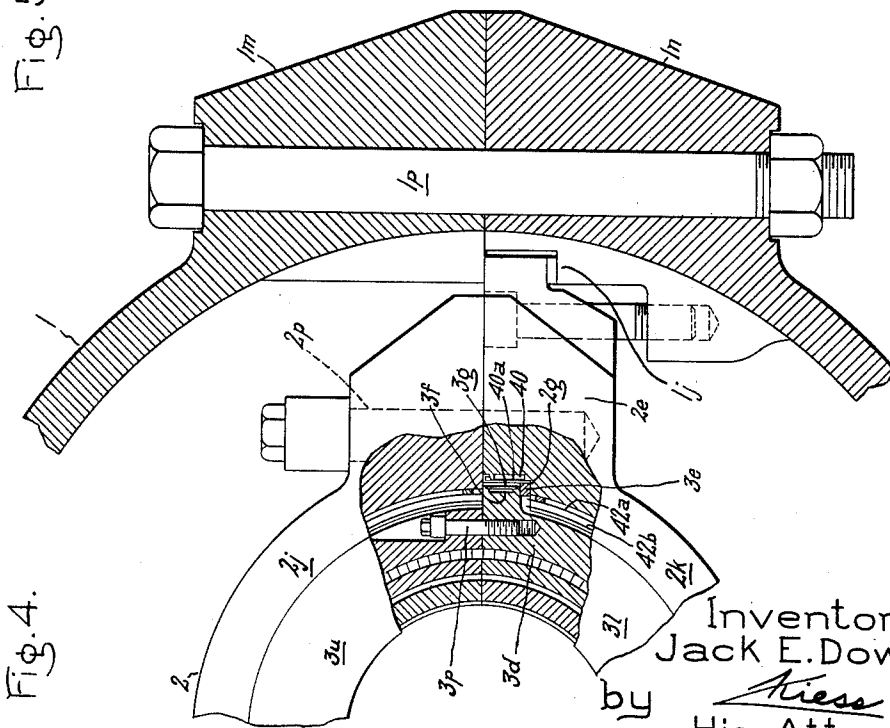


Fig. 4.

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2,815,645

SUPER-CRITICAL PRESSURE ELASTIC FLUID TURBINE

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Application March 1, 1955, Serial No. 491,290

14 Claims. (Cl. 60—40)

This invention relates to elastic fluid turbines for extremely high pressures and temperatures, specifically to turbines adapted for use with steam substantially above the critical pressure and temperature of water. The invention relates particularly to the mechanical construction which permits the use of such extremely high pressures and temperatures, and the special cooling system for the pressure-resisting parts.

As the size of turbine-generator units required by large electric utility generating systems has increased, there has been felt an ever greater necessity for converting the thermal energy in the fuel into electricity with the greatest possible efficiency, because of the enormous savings in fuel cost which may be effected in the course of a year by even a very small percentage increase in the thermal efficiency of the plant. This search for ever greater thermal efficiency has led to consideration of steam turbines operating with motive fluid above the "critical pressure" of water, which may be simply defined as the saturation pressure corresponding to the temperature above which there is no difference between the liquid and gaseous state of the fluid. For water, the critical temperature is 705.4° F. and the critical pressure is 3206.2 pounds per square inch, absolute (p. s. i. a.).

In order to obtain a worthwhile improvement in thermal efficiency of the steam turbine power plant, it has been ascertained that it is necessary to go to pressures on the order of 3500 to 5000 p. s. i. a. Specifically, the turbine for which the invention was developed is intended to operate with steam at a pressure of 4500 p. s. i. a. and an initial temperature of 1150° F.

For operation at such extreme pressures and temperatures, it is essential that the greatest care be given to the mechanical design of the turbine and the selection of appropriate materials. Even with the best materials available, it is necessary to design the apparatus to operate very close to the maximum allowable limits for the various materials. Accordingly, it becomes of utmost importance to insure that proper temperature conditions, and changes thereof, be insured in the normal operation of the power plant. Because of the close clearances to which the turbine must be manufactured, any distortion resulting from excessive temperature gradients may easily result in rubbing and severe damage to the machine.

In connection with similar high temperature apparatus, it has previously been noted that it is desirable to separate the "temperature-withstanding function" from the "pressure-withstanding function." The hot parts can then be made of temperature resisting material of thin section, having little or no pressure drop thereacross, while a suitable cooling system is provided for the "pressure-withstanding parts," which must of course be of heavier wall section and can, if the cooling is properly arranged, be of a lower temperature material. This is of course very important because high temperature alloy materials are expensive, difficult to work, have lower strength and a much higher coefficient of thermal expansion than do the lower temperature alloy steels customarily used in turbine

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construction. The high coefficient of thermal expansion makes it particularly difficult to arrange for free differential thermal expansion between associated parts without setting up excessive stresses or reducing below a safe minimum the operating clearances between relatively moving parts.

Accordingly, an object of the present invention is to provide an improved high temperature, high pressure elastic fluid turbine construction having thin high temperature members carrying the hot motive fluid and carefully arranged so as to be able to freely expand and contract, the heavier pressure resisting members being maintained at lower temperatures by special cooling schemes.

A further object is to provide improved means for maintaining the hot members as hot as possible, while effectively cooling and maintaining the pressure resisting members at a lower temperature.

Another object is to provide an improved high temperature high pressure conduit structure for steam at super-critical pressures.

A further object is to provide turbine construction of the type described in which the hot parts have very little or no pressure drop thereacross, and likewise little or no temperature gradient from the inside to the outside thereof, the cooler pressure-withstanding parts likewise having only a very small temperature gradient thereacross.

A still further object is to provide an improved cooling system for a very high temperature turbine in which a minimum amount of cooling fluid is required, and the pressure and temperature energy thereof is conserved by passing the used coolant through the energy-extracting bucket-wheels of the turbine rotor.

Other objects and advantages will become apparent from the following description, taken in connection with the accompanying drawings, in which

Fig. 1 is a longitudinal view, partly in section, of a super-critical pressure turbine incorporating the invention;

Fig. 2 is an enlarged detail view of the motive fluid inlet conduit and the highest pressure portion of the turbine;

Fig. 2a is an enlarged detail view of the assembly of the diaphragm members of Fig. 2;

Fig. 3 is a perspective view of a portion of the high pressure inner shell appropriately broken away to illustrate the path of the coolant flow;

Fig. 4 is a transverse sectional view, taken at the plane 4—4 in Fig. 1, showing the flanged joints between sections of the inner and outer casings; and

Fig. 5 is another transverse section, at the plane 5—5 in Fig. 1, illustrating further details of the casing structure.

Generally stated, the invention is practiced by making the high temperature members of thin wall section, arranged so as to be able to freely expand and contract with temperature changes and having a fluid pressure surrounding the members of the same general magnitude as the pressure within the members, insulating the high temperature members to reduce the temperature gradient thereacross, and providing the heavier pressure-resisting members with a cooling system which likewise reduces to a minimum the temperature gradient across these thick-walled members. Thus, the hot parts are maintained at their most efficient operating temperature, with no substantial temperature gradients through the wall sections to cause "thermal fatigue" failure of the parts, while the comparatively heavy pressure resisting parts are maintained at a temperature appropriate to the lower temperature material of which they are fabricated, and the elimination of temperature gradients also increases the reliability and life of these parts.

Referring now more particularly to Fig. 1, the invention is illustrated as applied to an extremely high pressure turbine comprising a generally spherical outer casing

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1, an intermediate inner shell assembly 2, and an inner high pressure assembly indicated generally at 3. The special inlet conduit assembly for conducting the motive fluid through the outer casing 1 and the intermediate casing 2 and into the inner casing assembly 3 is indicated generally at 4.

The outer casing 1 is made of generally spherical configuration, in order to effectively stand the pressure differential to which it is subjected. Since the pressure in the chamber 5 defined between the outer casing 1 and the intermediate casing 2 will be on the order of 1265 p. s. i. a., and the temperature about 800° F., the outer casing may be made of ordinary ferritic cast steel, which is comparatively inexpensive, easy to work, and readily capable of standing temperatures of this magnitude. It will be apparent from Fig. 1 that the casing is provided with suitable bearings 1a, 1b, and shaft packings 1c, 1d, 1e, 1f, 1g. These bearings and seals may be of appropriate conventional construction, the details of which are not material to an understanding of the present invention. It is of some importance here to note that the seals are provided with appropriate "leak-off" conduits identified 6a, 6b, 6c, 6d, 6e, 6f, 6g. The disposition of the leakage fluid discharged from these conduits will be noted as the description proceeds.

The turbine rotor, indicated generally at 7, has appropriate portions cooperating with the bearings and shaft seals, and has a plurality of bucket-wheels 7a . . . 7k secured to or formed integral therewith. Each bucket-wheel has a circumferential row of blade or bucket structures, forming the energy converting portion of the motive fluid flow path. Between each adjacent pair of bucket-wheels is an annular shaft packing member, two of which are identified 8a, 8b. The mechanical details of these packings will be understood by those skilled in the art and are not material to an understanding of the present invention.

The outer casing 1 is divided into an upper and lower half 1m, 1n secured together by a row of bolts 1p at a horizontal flange joint, the details of which may be seen in Figs. 4, 5. The intermediate shell 2 is likewise made of upper and lower casing halves 2j, 2k secured together by studs 2p.

The innermost shell or casing assembly 3 is composed of a plurality of semi-annular transverse partitions, or "diaphragm" members identified 3a, 3b, 3c, 3d, made of high temperature resisting austenitic alloys. Each of these elements comprises two 180° segments or sections suitably secured together by studs 3p at a diametral plane, as shown in Fig. 4. The sections 3a-3d are secured to each other to form the inner shell by a special annular "hook" arrangement, one of which is illustrated to an enlarged scale in Fig. 2a. It will be seen that the shell section 3d is provided with an axially extending arcuate portion 3e defining a circumferentially extending "tongue" 3f and a circumferential groove 3g. The adjacent shell section 3c has a portion defining an annular groove 3h and a tongue 3i. The tongue 3i and groove 3g define a small clearance space 3j, while the other pair of cooperating elements 3f, 3h form a similar clearance space 3k.

It will be obvious that in assembling the innermost shell it is necessary first to insert section 3a in the lower half of shell 2, then section 3b may be installed with the hook edge portions engaged as shown in Fig. 1. Succeeding sections 3c, 3d are similarly installed. Then, after the top inner casing 3u is lowered onto the lower casing half 3l (Fig. 4), the studs 3p may be inserted to hold the respective sections of diaphragms 3a, 3b, 3c, 3d together. When thus assembled, the clearance space 3j, 3k (Fig. 2a) permit the respective adjacent sections to expand and contract radially with temperature changes without imposing excessive stresses on the adjacent sections. It will be apparent to those skilled in the art that there will be a progressive drop in the temperature of the motive fluid as it passes from the inlet end of the

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innermost shell through the diaphragm sections 3a, 3b, 3c, 3d in series, so that each will operate at a lower temperature level. With this casing arrangement, each section can assume a temperature corresponding to that of the motive fluid passing through it, and can experience any change caused by changes in the temperature of the fluid, without creating excessive temperature gradients in any integral metal structures. The method of supporting the inner casing assembly 3 in the intermediate shell 2 will be seen more particularly hereinafter in connection with Figs. 3, 4, 5.

The intermediate shell 2 is supported in the outer casing 1 by suitable means, only a portion of which is illustrated in Fig. 1. This support includes a radially extending casing portion 2a having formed at the outer end thereof a radial key member 2b, of square or rectangular cross-section. The outer casing has a boss portion 9, provided (for manufacturing convenience) with a tubular insert member 9a having an inner end portion defining an axially extending slot 9b, adapted to receive the key 2b. The outer end of boss 9 may be closed by a suitable cover plate member 9c. It will be apparent that this slot and key arrangement centers the left-hand end of the shell 2 while permitting it to slide axially as temperature changes in the shell 2 cause it to expand and contract longitudinally.

At the right-hand end of shell 2 there is provided an analogous key and slot arrangement, including a radially extending groove 10a defined by a boss portion 10 formed on the end of casing 2. The cooperating radial key 10b is formed integral with or secured to an adjacent boss portion 10c of the outer casing 1. It will be obvious that, with the two halves of intermediate shell 2 bolted together, three or more key and groove arrangements similar to 2b, 9b, and 10a, 10b, respectively, would serve to center the intermediate shell 2 within the outer casing 1. However, in this turbine, the intermediate shell 2 is supported on suitable seat portions 1j defined in the outer casing at the horizontal flange joint, as may be seen in Fig. 4. With the intermediate shell thus supported at the horizontal plane, only one key and slot arrangement like those shown at 2b and 10a in Fig. 1 at either end of the casing are required, in order to maintain the intermediate shell 2 in proper coaxial relation with the outer casing.

It is also to be noted that the intermediate casing 2 is located longitudinally with respect to the outer casing by an interengaging circumferential rabbet portion located in a transverse plane in the neighborhood of the inlet conduit 4, as shown in Fig. 1 and in more detail in Fig. 5. Here it will be observed that the intermediate shell flanges define arcuate rib portions 2h, 2i defining a keyway receiving a key member 1r formed on an internal boss portion 1s at the horizontal flange joint of the outer casing 1. Thus, the intermediate casing 2 is fixed in the outer casing at its right-hand end, and is permitted to expand longitudinally to the left as the temperature of the casing increases.

The diaphragm assembly forming the second group of expansion stages in the casing 2 include annular nozzle diaphragms of conventional construction, identified 11a . . . 11f (Fig. 1). It will be appreciated by those familiar with the turbine art that each is formed in two halves bolted together at a horizontal plane through the axis of the machine and supported in circumferential grooves in the shell 2, as will be apparent in Fig. 1. By the time the motive fluid reaches this portion of the flow path, the temperature and pressure have been reduced to values such that types of construction previously used in the turbine art may be employed. Therefore, details of the construction of this portion of the turbine need not be described here.

The inlet conduit assembly 4 comprises a heavy outer conduit member 12, an intermediate spacer tube or shield member 13, and a high temperature inner conduit 14.

The details of this inlet conduit assembly may be better seen by reference to the enlarged view in Fig. 2.

The outer conduit member 12 is of heavy wall section to take the pressure forces, and may in some cases be made of ordinary ferritic steels (preferably a steel forging) since it is carefully protected from high temperatures, as described hereinafter. In the present design, it was actually fabricated of high temperature resisting austenitic-alloy. At its upper end, the member 12 is provided with a heavy circumferential flange 15, which is secured to a reinforced casing portion 16 by a plurality of studs 17. Interposed between the flange 15 and the nuts 17a is a heavy ring member 17b, the function of which is to distribute the force exerted by the nuts on the flange 15, at the same time permitting the use of long studs in order that the nuts can conveniently be tightened by heating the stud, tightening the nut, and permitting thermal shrinkage of the stud to increase the tightness of the joint. Such bolted flange constructions are common in the high temperature turbine art and do not form a part of the present invention. The upper end portion of conduit member 12 is welded at 12a to the adjacent end of the main steam supply conduit 18. As will be apparent from Fig. 2, the end portion of conduit 18 defines an inner terminal portion 18a welded to the upper end of the inner high temperature conduit 14. It will also be apparent that the heavy outer conduit member 12 and the inner high temperature conduit 14 define an annular chamber in which is disposed the special spacer member 13.

The function of this spacer 13 is twofold. In the first place, it will be observed that the inner surface of member 13 is provided with a plurality of axially spaced concentric ribs, as shown in dotted lines at 19. These parallel annular ribs define a plurality of insulating "dead steam" spaces 19a, the function of which is to reduce the transfer of heat from the hot thin-walled conduit 14, thus reducing to a minimum the temperature gradient through the thin wall of conduit 14. In the second place, the outer surface of spacer 13 is provided with a helical rib 20, forming a helical flow path 20a for the cooling steam, which is admitted in the following manner.

By reference to Fig. 1 it will be seen that the flange 15 is provided at one location in its circumference with a radially extending passage 15a, to which is supplied comparatively cool steam by a coolant supply conduit 21, which has an enlarged end portion 21a welded to the flange 15, in a manner which will be apparent from Fig. 2. Here it will be seen that the conduit end portion 21a has welded thereto, or formed integrally therewith, a thin-walled tube 21b which projects into the radial passage in flange 15 and defines therewith a small clearance space 22, in order to reduce the transfer of heat from the hot flange 15 to the cooler conduit 21. The extreme inner end of conduit 21b is provided with an enlarged portion 21c, which does contact the bore in flange 15 to maintain the annular clearance space 22. It is important to thus reduce the transfer of heat between the coolant supply conduit 21 and the flange 15 in order that the incoming coolant will not cause a local "cold spot" in the massive flange 15. It will be obvious that any unbalance of the circumferential temperature distribution in the massive flange 15 may produce a tendency to crack.

The entire design of this turbine is carefully arranged to keep all temperature distributions symmetrical in the respective parts affected. To further this purpose, the inlet conduit 21b discharges the cooling fluid into an annular groove 23 formed in the inner bore of flange 15. This annular groove serves as a distributing chamber to feed coolant uniformly around the spacer member 13 into the helical cooling passage 20a. The use of the helical cooling passage 20a further insures that the circumferential temperature distribution around the heavy conduit member 12 will be uniform. It will be appreciated by those skilled in the art that it would be extremely diffi-

cult, if not impossible, to provide parallel longitudinal cooling flow paths between the hot inner conduit 14 and the cooler outer conduit 12 and insure that coolant would be supplied uniformly to all of these parallel paths. By using a single helical flow path, complete uniformity of temperature distribution around the circumference of the hot conduit is insured. It is of course true that the temperature of the coolant will progressively increase as it passes downward along the spiral path 20a. The thing to be avoided is an uneven circumferential temperature distribution at a given transverse plane through the conduit assembly.

The lower end portion of conduit 12 is received in a boss portion 2c of the intermediate shell 2, and the leakage of steam is prevented by a multiple ring slip-joint indicated generally at 2d. The mechanical details of this joint are not necessary here, but it may be of the construction illustrated in the patent of P. G. Ipsen, No. 2,649,315, issued August 18, 1953, and assigned to the same assignee as the present application. It will be appreciated by those skilled in the art that this joint permits differential thermal expansion in a radial direction between the conduit 12 and the boss portion 2c, at the same time permitting limited transverse displacement, by reason of slippage between the end surfaces of the packing rings in a direction transverse to the axis of the inlet conduit, as is necessary to take care of differential thermal expansion in a longitudinal direction between the outer casing 1 and the intermediate shell 2.

As will further be seen by reference to Fig. 2, the lower end of the high temperature pipe 14 is enlarged at 24 to receive the adjacent end portion 25 of an inlet conduit formed integral with or secured to the nozzle-box 26. The nozzle box 26 extends circumferentially approximately 90° so as to feed the nozzles 27 around approximately a quarter of the circumference of the bucket wheel 7a.

It will be appreciated that there will be a plurality of these inlet conduit and nozzle-box assemblies. In the present case there are four, two in each casing half, equally spaced around the circumference of the casing.

The nozzle-box 26 is surrounded by a spaced box member 28, which is supported at its upper end by a suitable rabbet at 28a in the shell 2, having another rabbet fit with the nozzle box at 28b adjacent the nozzles 27.

As will be apparent from Fig. 2, the shaft packing members 1e are supported in an annular shell member 29 which is suitably supported by a rabbet in shell 2. The radially inward thrust on the nozzle box member 26, due to the pressure of the fluid inside, may be transmitted through a boss 26a to the bottom wall of box member 28, thence by way of a boss 28c to the outer surface of the shell member 29. It will be observed that the box 28 is spaced from the adjacent portions of packing shell 29 and casing shell 2 to define a coolant flow passage 30. Disposed between the nozzle-box 26 and the box 28 is a thin shield member 31 forming an outer insulating "dead space" 31a and an inner substantially stagnant space 31b.

Still referring to Fig. 2, the upper end of nozzle-box supply conduit 25 is disposed within the enlarged end portion 24 of conduit 14, and leakage therebetween is prevented by a multiple ring slip-joint indicated generally at 32. This joint may be of generally similar construction to the joint 2d. The conduit end portion 24 is surrounded by a cylindrical shield member 33 which is seated in a recess in shell 2. The central opening in the bottom of member 33 receives the upper end portion of a cooperating shield member 34 which may be formed integral with or secured to the upper end portion of box member 28. The upper end portion of shield 33 surrounds the enlarged end portion 24 of conduit 14, leakage therebetween being resisted by a pair of sealing rings 35. It remains to note that the lower end of the intermediate spacer member 13 is provided with a flaring end portion 13a, the extreme lower end of which surrounds and forms a small clearance space with the adjacent end portion of the cylindri-

cal shield 33. These shield members serve to define the cooling flow path as described more particularly herein-after.

The supply of cooling steam may be obtained from any suitable source, for instance from any appropriate point in the steam generator where the temperature and pressure are at the correct values. It may conveniently be taken from the main steam generator supply pipe and cooled by "desuperheating" by injection of water. It need only be noted here that cooling steam is supplied to the conduit 21 at a temperature on the order of 960° F. and at a pressure slightly above the "first stage shell pressure," which is that existing at the downstream side of the first bucket-wheel 7a. Specifically, this pressure may be on the order of 3900 p. s. i. a., as compared with a first stage shell pressure of 3800 p. s. i. a. The supply of cooling steam is controlled by means represented diagrammatically by the valve 21e in Fig. 1. The rate of cooling steam flow is further limited by an orifice plate represented at 21f. The valve 21e will of course be adjusted to provide the minimum quantity of cooling steam adequate to perform the temperature control functions required, since this cooling steam flow represents a thermal loss to the power-plant. It will be seen as the description progresses how the thermal energy picked up by this cooling steam is eventually utilized, so that it does not represent a total loss.

The flow of cooling steam in the inlet conduit assembly is as follows, referring to Fig. 2.

The cooling steam is admitted through conduit portion 21b to the annular supply groove 23 in the flange 15 and fed into the helical flow path 20a. The major flow will of course be downward along path 20a, but it is to be noted that this cooling steam has access around the upper end of the spacer member 13 to the annular dead spaces 19a through the small clearances with conduit 14 defined by the annular lands 19. This access to the chambers 19a is represented by the arrow 20b at the upper end of spacer 13. At the lower end of spacer 13, the cooling flow is diverted outwardly by the flaring portion 13a into the annular space 33b defined between the shield 33 and the shell 2. From this chamber the coolant flows by one or more passages 36 in shell 2 to an annular chamber 34a defined in the upper end wall of box member 28. From this supply annulus 34a, coolant flows by way of passages 37 drilled through the upper end portion of box 28 into the chamber 30. The lower left-hand portion of box 28 carries an annular shield member 28d which directs the flow of coolant from chamber 30 radially inward across the end surface of shaft seal support shell 29. Some of this coolant leaks outwardly to the right past the shaft seal segments 1e, providing the required cooling for these sealing members. From the right-hand end of shaft packing 1e, spent coolant is discharged through the conduit 6d. The balance of this portion of the cooling flow flows radially outward between the shield 28d and the first stage bucket-wheel 7a and passes axially through the wheel by way of a plurality of holes identified 7m. The coolant then flows radially inwardly along the downstream side of the wheel 7a, leaks through the first shaft packing 8a, then flows radially outward and through axial passages in the second stage wheel 7b, etc. This coolant continues through the shaft packing 8b, and through similar axial passages in the wheel 7c, then through the packing 8c and through the wheel 7d and packing 8d, as shown by the flow arrows in Fig. 2.

It is to be noted that the first four bucket wheels, 7a . . . 7d are provided with circumferential shoulders defining annular "spill strips," as identified at 7n in Fig. 2. These annular projections are carefully designed to form close radial clearances with adjacent circumferential portions of the respective diaphragms so as to restrict to a minimum the leakage of cooling steam into the main stream of hot motive fluid flowing through the buckets and nozzles. Thus, the coolant is confined to the leak-

age path along the rotor and through the bucket-wheels as described above. A similar cooling system for a high temperature steam turbine rotor is described more particularly in the patent of G. B. Warren, No. 2,552,239, issued May 8, 1951, and assigned to the same assignee as the present application. By the time this coolant reaches the bucket-wheel 7e it has picked up enough heat to have little further effect as a coolant; and, by the same token, the temperature of the motive fluid has decreased so that wheel 7e and subsequent wheels do not need this cooling flow. Accordingly, there are no spill strips, corresponding to the projections 7n, on the bucket-wheel 7e, the coolant being permitted to mix at this point with the motive fluid as indicated by the flow arrows. Thus some of the energy in the spent coolant is delivered to the subsequent bucket-wheels, reducing the net loss chargeable to the cooling system.

The design of the cooling system for the inlet conduit assembly is carefully arranged so that there will be no accidental leakage of motive fluid outwardly into the coolant. Specifically, any leakage of hot motive fluid from the conduit 14 past the sealing rings 32 will flow downwardly between the nozzle box inlet conduit 25 and the shield member 34 and through a port 34b into a chamber 34c defined between the upper surface of nozzle box 26 and the adjacent portion of shell 2. This chamber is in free communication with the chamber containing the first stage buckets, so that any hot fluid entering chamber 34c will be directly introduced into the nozzle passages of the first diaphragm section 3a.

It may also be observed that coolant entering the clearance space between the flaring portion 13a of spacer 13 and the enlarged end portion 24 of pipe 14 will have access to the dead spaces 19a defined between the parallel annular lands 19. It will be appreciated that these lands 19 are not intended to form a fluid-tight fit with the outer surface of pipe 14, so that the small clearance spaces therebetween will permit equalization of the fluid pressure in the respective "dead" spaces 19a. In normal operation the pressure in these insulating spaces will be substantially equal to that of the coolant in the helical flow path 20a. Thus, since there is no pressure drop across the spacer member 13, it can be made of very thin wall section.

The construction and method of operation of the cooling system for the shell 2 associated with the high temperature diaphragm assembly 3a . . . 3d will be seen from the following.

Coolant from the annular supply chamber 34a in Fig. 2 is supplied by way of a port 34d in the upper end portion of box member 28 to a chamber 38 defined between adjacent portions of box 28 and the shell 2, and thence by way of a passage 39 in shell 2 into a longitudinally extending passage 40. The quantity of coolant flowing, and the pressure thereof, is determined by an orifice plate 38a, which may be conveniently inserted at the entrance to passage 39. This is designed to effect a pressure in the passage 40 on the order of 3850 p. s. i. a.

Attention is directed to the fact that the longitudinally extending passage 40 in shell 2 is actually located at the position of the horizontal flange joint which connects the halves of intermediate shell 2. In order to be able to show all portions of the cooling flow path in one figure, the passage 40 has been rotated into the vertical plane in Fig. 2. Its actual location will be more apparent from a consideration of the perspective view in Fig. 3.

Fig. 3 shows a broken-away view of a portion of the horizontal flange joint of the intermediate shell 2, including a portion of the upper shell 2j, and the lower shell 2k, the separation plane between which is identified 2f. This view also shows a portion of a projecting flange portion of diaphragm section 3d by which the diaphragm is supported on the lower shell half 2k. This support means includes a rectangular key member 3e

interposed between the lower surface of diaphragm flange 3d and a horizontal seat 2g formed on the lower shell section 2k (see also Fig. 4). In assembling this arrangement, a temporary key member like 3e will be inserted and the diaphragm section 3d installed and aligned with the intermediate shell 2. Then the key member which is actually to be used is machined to such dimensions that when installed as shown at 3e in Figs. 3 and 4 the diaphragm section 3d will be supported in proper coaxial relation with the shell 2.

Both upper and lower sections of the intermediate shell 2 are provided with chambers corresponding to that identified 38 in Fig. 2, with passages corresponding to that identified 39. This coolant from the upper shell section is admitted through passage 39a in Fig. 3, so that the coolant derived from both the upper and lower shell sections is admitted to the same longitudinal coolant passage 40.

By comparison of Figs. 2 and 3, it will be seen that the channel 40 is defined by a longitudinal groove in the lower flange member 2e of shell 2, which groove is covered by a thin plate member 40a. It will also be noted that the diaphragm flange member 3d is similarly provided with a longitudinal groove 3f covered by a plate member 3g. The resulting chamber 3f is not a coolant passage but a dead or stagnant space for heat insulation, preventing the transfer of heat from the comparatively hot flange 3d to the coolant fluid in passage 40. It will also be apparent from Fig. 3 that the two cover plate members 40a and 3g are spaced apart to define a further stagnant insulating space 3h. Thus, there is a double barrier to the transfer of heat from the hot flange 3d to the coolant fluid in passage 40.

Coolant from the extreme left-hand end of passage 40 is admitted by a port 40b to a vertical passage 40c drilled in cooperating portions of both the upper and lower shell halves 2j, 2k. The lower end of passage 40c communicates with a drilled hole 40d which supplies coolant to a 90° segment of the lower casing half. The upper end of passage 40c conducts coolant to similar passages in the upper shell 2j.

The path by which the coolant supplied through passage 40 returns along the outside of the hot diaphragm sections 3a . . . 3d will be seen from a comparison of Figs. 2 and 3. In Fig. 2 it will be seen that an intermediate shield support casing portion 41 has a plurality of axially spaced inwardly extending circumferential ribs 41a, 41b, etc. These ribs have arcuate grooves receiving the respective edge portions of a pair of parallel radiation shield members 42a, 42b. These shields define an arcuate stagnant insulating space 42c. Another substantially stagnant space 42d is defined between the inner plate 42b and the adjacent outer surface of the diaphragm section 3d (Fig. 2). The path for the coolant is defined between the outer shield 42a and the shield support member 41 and is identified 42e in Fig. 2. It will be seen that here also the coolant passage 42e is separated by two stagnant heat insulating chambers 42c, 42d from the hot diaphragm member 3d.

In Fig. 3 it will be apparent that coolant is admitted through passage 40d into the outer arcuate passage 42e, in which it flows circumferentially as indicated by arrows 42f to a point adjacent the bottom center of the diaphragm section 3d. At this location, the circumferential rib 41a is provided with one or more ports 42g, which admit the coolant to the corresponding arcuate passage 43e defined by the outer radiation shield member 43a. In this passage the coolant flows back upward toward the horizontal flange joint, as indicated by the arrows 43f. At the upper end of passage 43e, the circumferential rib 41b is provided with one or more ports 43g which admit the coolant to the next arcuate passage 44e. Here, the coolant again flows downwardly to the middle bottom, then through axial ports into the next arcuate passage 45e (Fig. 2), and similarly through ports in the succeed-

ing ribs 41c, 41d, 41e and the respective arcuate passages 45e, 46e, 47e, 48e.

In order to return the spent coolant to the motive fluid flow path of the turbine, the last pair of radiation shields 48a, 48b are provided with a series of ports 49 which admit the coolant to the annular chamber 34c (Fig. 2), whence it enters the motive fluid flow path in the first diaphragm 3a, as indicated by the arrows in Fig. 2. Thus, the thermal energy picked up by the coolant is returned to the working fluid path of the turbine so that some use is made of this energy, also.

It will be seen that the lowest temperature coolant is supplied to the low temperature end of the cooling passages in the inner shell assembly 3, so that the coolant flows in "counter-flow" relation to the motive fluid.

It should be noted that the design of the cooling passages, and the size of the metering orifice 38a, are such that the pressure of the coolant to which the outer surface of the diaphragm sections 3a—3d are subjected, will tend to hold the respective diaphragm halves radially inward, so as to firmly engage the interlocking hooked edge portion (refer to 3f, 3i in Fig. 2a). In this connection, it will be noted from Fig. 2 that spent coolant from ports 49 has free access to the essentially stagnant spaces 42d, 43d . . . 48d defined between the inner radiation shields 42b, 43b . . . 48b and the outer surfaces of the diaphragm section 3a—3d. Thus the same fluid pressure is applied to the outer surfaces of all the diaphragm section 3a . . . 3d, although it is not intended that there be any particular flow of coolant in this stagnant space. It will be appreciated that, since this coolant is at a pressure slightly above that existing at the entrance to the first diaphragm section 3a, there will be a substantial pressure differential on the diaphragm sections tending to hold them together. This also means that, if there should be any leakage path between a pair of adjacent diaphragm sections, this leakage would be of coolant fluid inwardly rather than hot motive fluid outwardly. This further tends to prevent the transfer of heat from the hot motive fluid flow path radially outwardly towards the cooler casing 2.

It will be seen that, both in the design of the inlet conduit 4 and in the intermediate shell 2, the cooling system is so arranged that the inner hot portions are surrounded by a heat insulating stagnant space which tends to maintain the hot metal walls at a high temperature, so there will be no more than a very small temperature gradient through the comparatively thin temperature-resisting walls. These insulating spaces also prevent the transfer of an excessive amount of thermal energy outwardly into the coolant flow path. The flow of coolant which surrounds these insulating spaces is not intended to cool the hot members, but is instead intended to carry away any heat which leaks outwardly from the hot members past the thermal insulating spaces, before it has a chance to reach the heavy outer walls which perform the pressure-withstanding function.

It will be apparent from Fig. 2 that the outer conduit member 12 and the intermediate shell 2 are both surrounded by spent motive fluid, which in normal operation will be at a temperature on the order of 800° F., while the inner surfaces of these heavy members are blanketed by the cooling steam, at a temperature on the order of 960° F. Thus, the temperature differential through the heavy pressure-resisting walls of inlet conduit 12 and shell 2 may be a maximum of perhaps only 160° F. By thus reducing the temperature gradient occurring in these heavy metal sections, it is possible to greatly prolong their life, and use the lower cost ferritic alloys, which are quite serviceable at such temperatures.

It will also be observed that the comparatively cool steam in the chamber 5 circulates around the shell boss portions 2c and radially inwardly between the right-hand end of shell 2 and the adjacent outer casing portion 1h so as to furnish comparatively cool steam to the leakage

path through the shaft seal 1f, as indicated by arrow 50 in Fig. 1. This leakage is necessary in order to keep the shaft seal elements cool. Some of this cooling leakage is drawn off through the conduit 6e as indicated by arrow 51, the remainder leaking through the last shaft seal section and escaping through passage 6f as indicated by arrow 52.

Similarly, spent motive fluid leaks to the left through the shaft seal 1d, some of this leakage escaping through conduit 6c while the rest flows through the last shaft seal segment and escapes through conduit 6b, as indicated by the flow arrows at the left-hand side of Fig. 1. This comparatively cool steam withdrawn through conduits 6c, 6e may be disposed of to a common device for making use of steam at such temperatures and pressures, for instance a feedwater heater, or to suitable lower pressure stages of another turbine. Likewise, the leakage from conduits 6b, 6f may be combined and disposed of in a feedwater heater or other appropriate consumer. On the other hand, the shaft seals 1c, 1g will have steam leaking in from one side and air drawn in from the external side. Thus, the conduits 6a, 6g will be connected to a suitable condenser (not shown) maintained at a vacuum to draw off this mixture of air and steam.

It may also be noted that the comparatively high temperature cooling steam leakage past the shaft seal 1e is withdrawn through the conduit 6d so it cannot reach the subsequent shaft packings 1f, 1g and cause them to overheat. This hot leakage withdrawn through conduit 6d may, for instance, be disposed of in the manner disclosed by the patent of C. W. Elston, No. 2,467,818; issued April 19, 1949, and assigned to the same assignee as the present application.

It will be seen that the invention provides a novel high temperature turbine for extremely high pressure operation in which the pressure-withstanding function is carefully divorced from the temperature-resisting function. The hot parts are made of very thin-walled sections of high temperature resisting material having substantially no pressure drop thereacross. Furthermore, the hot members are provided with heat insulation means preventing the loss of heat therefrom, so as to reduce to a minimum the temperature gradient existing through the hot walls. The outer pressure-resisting walls are of much heavier section and fabricated of lower temperature alloys, and are provided with a cooling system which keeps to a minimum the temperature differential existing across these heavy walls. Thus, the cooling system is carefully designed to perform the dual function of reducing loss of heat from the hot members and reducing transfer of heat to the cold members. By reducing or eliminating temperature gradients across both the hot members and the colder pressure-withstanding members, the tendency to form thermal fatigue cracks is reduced, and the design problems resulting from differential thermal expansion between cooperating parts are simplified. Another way to state this design theory is that the temperature drops are taken across stagnant heat insulating spaces rather than across a metal wall, while the pressure drop is taken across a thick metal wall provided with a cooling system so that again there is little or no temperature differential thereacross. This design keeps to a minimum the number and mass of parts which must be made of expensive, hard-to-fabricate, austenitic alloys, thus further reducing the design problems which otherwise result from the comparatively greater differential thermal expansion of the austenitic alloys.

Certain modifications, both in the cooling flow path and in the mechanical design of the hot and cold parts will be apparent to those skilled in the art, and it is of course desired to cover by the appended claims all such changes as fall within the true spirit and scope of the invention.

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

1. In high temperature high pressure fluid pressure ap-

paratus, the combination of a first inner member having walls defining a first flow path for high pressure high temperature fluid and fabricated of high temperature resisting materials of comparatively thin wall section, a second shield member surrounding said first member and defining therewith second substantially stagnant thermal insulating spaces for reducing the transfer of heat outward from said first member, and a third pressure-resisting member fabricated of material having good strength properties at lower temperatures and of comparatively thick wall section surrounding said shield member and defining therewith a third path for the flow of cooling fluid at a pressure substantially that of the hot fluid within said first member, passage means providing limited access of cooling fluid from said third path to said second thermal insulating spaces to substantially equalize the pressure differential across the thin wall of said first member, whereby the pressure differential between the high pressure hot fluid inside the first member and ambient pressure surrounding said third member is taken across said third thick-walled member while the temperature drop occurs across said second insulating spaces with substantially no temperature gradient through the thin wall of the first member, the flow of coolant in said third path serving to prevent the transmission of heat from the hot first member to the cooler third member to prevent substantial temperature gradients in said thick-walled third member.

2. In high temperature high pressure fluid pressure apparatus, the combination of a first inner member having walls defining a first flow path for high pressure high temperature fluid, said first member being fabricated of high temperature resisting materials of comparatively thin wall section, a second shield member surrounding the first member and defining therewith a second substantially stagnant thermal insulating space for reducing the transfer of heat outward from the first member, and a third pressure-resisting member of comparatively thick wall section surrounding said shield member and defining therewith a third path for the flow of cooling fluid at a pressure on the order of magnitude of that of the hot fluid within said first member, means for supplying fluid to the second thermal insulating space at substantially the pressure of the hot fluid within said first member to substantially equalize the pressure differential across the thin wall of said first member, whereby the overall pressure differential is supported substantially by the third thick-walled member while the temperature differential occurs across the second insulating space, with a minimum temperature gradient through the thin wall of the first member, the flow of cooling fluid in said third passage serving to prevent substantial temperature gradients in said thick-walled third member.

3. In a high temperature high pressure fluid pressure apparatus, the combination of a first inner member having walls defining a first flow path for high pressure fluid and fabricated of high temperature resisting materials; a second shield member surrounding said first member and defining therewith at least one substantially stagnant thermal insulating space for reducing the transfer of heat outward from said first member, and a third pressure-resisting member of comparatively thick wall section surrounding said shield member and defining therewith a third path for the flow of cooling fluid at a pressure on the order of magnitude of that of the hot fluid within the first member, means for supplying fluid to said thermal insulating space at a pressure substantially that of the high pressure fluid within the first member to substantially equalize the pressure differential across the first member whereby the pressure differential between the hot fluid within the first member and the ambient atmosphere surrounding the third member is substantially taken across the third thick-walled member, while the temperature drop occurs substantially across the second insulating space with a minimum temperature gradient occurring through the wall of the first member, the flow

of coolant in said third path serving to prevent substantial temperature gradients in the thick-walled third member.

4. In a high temperature turbine casing assembly, the combination of a first thick-walled pressure-resisting casing having a discharge end portion supporting a plurality of diaphragm members defining intermediate pressure expansion stages, the first casing having an inlet end portion supporting an inner high temperature diaphragm assembly comprising a plurality of diaphragm members defining the initial high temperature stages of expansion and fabricated of high temperature resisting materials, a radiation shield assembly surrounding said high temperature diaphragms and defining with the outer surfaces of the diaphragms at least one stagnant thermal insulating space, said radiation shield assembly also having walls defining a second coolant flow path surrounding said insulating space, and means for supplying a coolant fluid at a pressure substantially equal to that obtaining at the inlet to said high temperature diaphragm assembly, walls defining a passage communicating said coolant to said insulating space, whereby any leakage between adjacent high temperature diaphragms will be of coolant fluid inwardly rather than hot motive fluid outwardly, said insulating space resisting the transfer of heat outwardly from the high temperature diaphragm sections, while the coolant flow in said second flow path keeps to a minimum the temperature gradient through said thick-walled first casing.

5. In a high temperature turbine casing assembly, the combination of a first thick-walled pressure-resisting casing fabricated of material having suitable strength at elevated temperatures and having an inlet portion supporting an inner high temperature diaphragm assembly comprising a plurality of diaphragm members defining high temperature stages of expansion and fabricated of suitable temperature-resisting alloys, a shield assembly surrounding said high temperature diaphragms and defining with the outer surfaces thereof at least one substantially stagnant thermal insulating space, said shield assembly also having walls defining a second coolant flow path surrounding said insulating space, and means for supplying to said second flow path a coolant at a pressure at least slightly greater than that obtaining at the inlet to the high temperature diaphragm assembly, walls defining a passage communicating said coolant pressure to said insulating space whereby any leakage between adjacent high temperature diaphragms will be of coolant inwardly rather than hot motive fluid outwardly, the insulating space resisting the transfer of heat outwardly from the high temperature diaphragms while the coolant keeps to a minimum the temperature gradient through the thick-walled first casing.

6. A high temperature turbine casing assembly comprising a first thick-walled casing having an inlet portion supporting an inner high temperature diaphragm assembly including a plurality of diaphragm members defining high temperature stages of expansion and fabricated of high temperature resisting materials, a shield assembly supported in said first casing and surrounding said high temperature diaphragm assembly and defining with the outer surfaces of the diaphragms at least one substantially stagnant thermal insulating space, the shield assembly also having walls defining a second coolant flow path surrounding said insulating space, means for supplying to said flow path a coolant fluid at a pressure at least slightly above the pressure of the motive fluid entering the high temperature diaphragm assembly, walls defining a passage communicating said coolant pressure to said insulating space whereby any leakage between the high temperature diaphragms will be of coolant inwardly rather than motive fluid outwardly, and walls defining a passage communicating spent coolant to the motive fluid inlet of said high temperature diaphragm assembly.

7. A high temperature turbine casing assembly in ac-

cordance with claim 6 in which the shield assembly comprises a cylindrical member with a plurality of axially spaced circumferential inwardly extending ribs, a plurality of arcuate plate members each disposed axially and supported between adjacent pairs of said ribs to define at least one arcuate coolant passage, the first thick-walled casing defining a supply passage for admitting coolant to one end of each of said arcuate passages at the discharge end of the high temperature diaphragm section, the rib at the other end of each of said arcuate passages defining at least one port for admitting coolant to the next adjacent arcuate passage, the next adjacent rib remote from said first arcuate passages defining a port for admitting coolant to the next arcuate passage, whereby coolant flows in succession in opposite directions through the arcuate passages formed by the plate members of the shield assembly.

8. A high temperature turbine casing assembly comprising a first thick-walled casing fabricated of materials having suitable strength at comparatively low temperature and having a portion supporting an inner high temperature diaphragm assembly including a plurality of diaphragm members defining high temperature stages of expansion and fabricated of high temperature resisting materials, a shield assembly supported in said first casing around said high temperature diaphragm assembly, the shield assembly being spaced from the outer surfaces of the diaphragms to form a substantially stagnant thermal insulating space, the shield assembly also having walls defining a coolant flow path surrounding said insulating space, means for supplying to said flow path a coolant fluid under pressure, walls defining a passage communicating the pressure of said coolant fluid to said insulating space, and walls defining a passage admitting spent coolant into the motive fluid flow path of the high temperature diaphragm assembly, whereby the insulating space reduces the transfer of heat from the hot diaphragm sections and the coolant flow reduces the temperature gradient in the thick-walled casing.

9. A high temperature turbine casing assembly in accordance with claim 8 in which the shield assembly comprises a cylindrical member with a plurality of axially spaced circumferential inwardly extending ribs, a plurality of arcuate plate members disposed axially and supported between each adjacent pair of said ribs to define at least one thermal insulating space surrounded by an arcuate coolant passage, walls defining a passage for admitting a cooling fluid to one end of the arcuate coolant passage located at the extreme discharge end of the high temperature diaphragm assembly, said circumferential ribs defining ports for causing the cooling fluid to circulate in succession in opposite directions circumferentially in adjacent arcuate coolant passages, whereby cooling fluid flows in succession in opposite directions through the adjacent arcuate coolant passages formed by the arcuate plate members of the shield assembly.

10. In a high temperature high pressure fluid conduit assembly, the combination of a first inner conduit member having walls defining a first flow path for high pressure high temperature fluid, said first member being fabricated of high temperature resisting material of comparatively thin wall section, a second shield member surrounding said first member and defining therewith a substantially stagnant thermal insulating space for reducing the transfer of heat outward from said first member, and a third cylindrical pressure-resisting member of comparatively thick wall section surrounding said shield member and defining therewith a third path for the flow of cooling fluid at a pressure on the order of that of the hot fluid within said first member, means for supplying fluid at a pressure substantially that of the hot fluid within said first member to said second thermal insulating space to substantially equalize the pressure drop across said thin-walled first member, whereby the pressure differential between the fluid inside the first member and ambient pres-

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sure surrounding said third member is taken across said third thick-walled member, while the temperature drop occurs across said insulating space with substantially no temperature gradient through the thin wall of the first member, the flow of coolant in said third path reducing the transmission of heat from the hot inner member to the third member to minimize temperature gradient in said thick-walled third member.

11. In a high temperature high pressure fluid conduit assembly, the combination of a first inner conduit member having comparatively thin walls of high temperature resisting material defining a first flow path for high pressure high temperature fluid, a second shield member surrounding said first member and defining therewith a second substantially stagnant thermal insulating space for reducing the transfer of heat from said first member, and a third pressure-resisting member of comparatively thick wall section surrounding said shield member and defining therewith a third path for the flow of cooling fluid at a pressure of the same order of magnitude as that of the hot fluid within the first member, said third member having a thick radially extending flange portion defining a coolant inlet passage, and means for supplying cooling fluid to said second space including a fluid inlet conduit member connected to said flange and having a second cylindrical shield member projecting into said coolant inlet passage, the major portion of said second shield member being spaced radially from the wall of the coolant inlet passage in said flange to define a thermal insulating space therewith, said thick-walled member defining an annular chamber surrounding the first conduit and having a portion communicating with said coolant inlet passage and communicating also with said third coolant path for supplying coolant fluid uniformly to said third path.

12. A high temperature high pressure fluid conduit assembly in accordance with claim 11, in which the inner conduit is cylindrical and the shield member comprises a cylindrical member surrounding and coaxially spaced from the first conduit, the shield member having a plurality of annular ribs projecting inwardly to substantially contact the outer wall of said inner conduit to form the stagnant insulating spaces, the cooling fluid path being defined by a helical rib in the space between said shield member and the thick-walled outer member, whereby the flow of coolant helically around the shield member provides uniform circumferential distribution of temperatures in said thick-walled outer member.

13. In a high temperature high pressure fluid conduit assembly, the combination of a first inner conduit member having comparatively thin walls of high temperature

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resisting material defining a first flow path for high pressure high temperature fluid, a second cylindrical shield member surrounding said first member and coaxially spaced therefrom, a third pressure resisting member of comparatively thick wall section surrounding said shield member and spaced therefrom, said shield member having a plurality of axially spaced annular ribs projecting inwardly to substantially contact the outer wall of the inner conduit to form substantially stagnant thermal insulating spaces, said shield member having also at least one external helical rib cooperating with the thick walled outer member to define a helical flow path extending along the inner conduit, and means for supplying cooling fluid to said helical flow path at a pressure of the same order of magnitude as that of the hot fluid within the first conduit member, and passage means communicating coolant fluid to the stagnant insulating chambers to minimize the pressure drop across the thin wall of the inner conduit member, the flow of coolant fluid in said helical path reducing the transfer of heat from the hot inner conduit member to the thick-walled outer member.

14. In a high temperature pressure fluid conduit assembly, the combination of a first inner conduit member having comparatively thin walls of temperature resisting material and defining a first flow path for high temperature fluid, a second cylindrical shield member surrounding said first member and defining therewith at least one substantially stagnant thermal insulating space for reducing the transfer of heat from the first member, and a third pressure-resisting member surrounding said shield member and defining therewith a third coolant fluid path, the third member having a radially extending flange portion defining a coolant inlet passage, means for supplying cooling fluid to said inlet passage at a pressure of the same order of magnitude as that of the hot fluid within the first member, the third member defining an annular chamber surrounding the first conduit and communicating between said stagnant insulating space and the coolant inlet passage and connected to supply coolant to said third path.

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