May 13, 1952

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RECUPERATIVE HEAT EXCHANGER OF THE COUNTERFLOW TYPE FOR GASEOUS MEDIA

2,596,622

Filed Jan. 18, 1947

7 Sheets-Sheet 1

Fig.1.

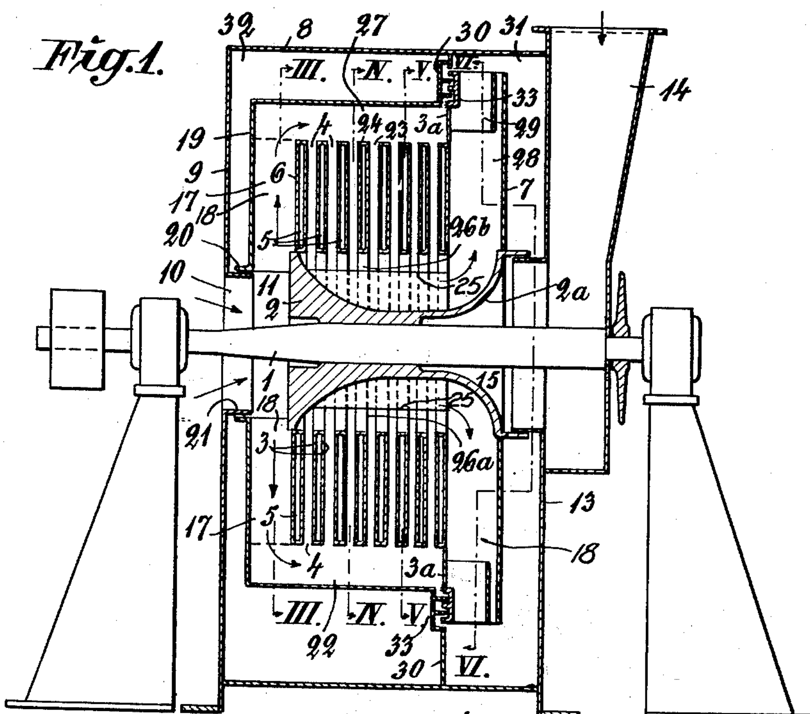

Fig.2.

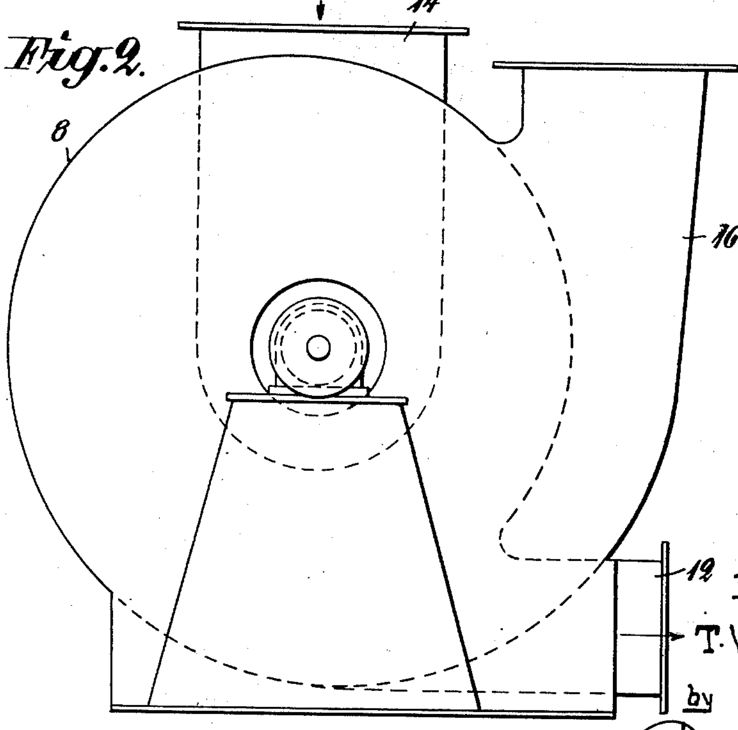

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Fig. 3.

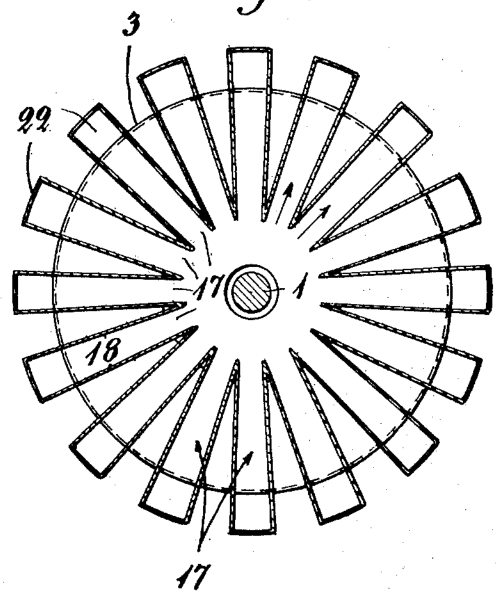

Fig. 4.

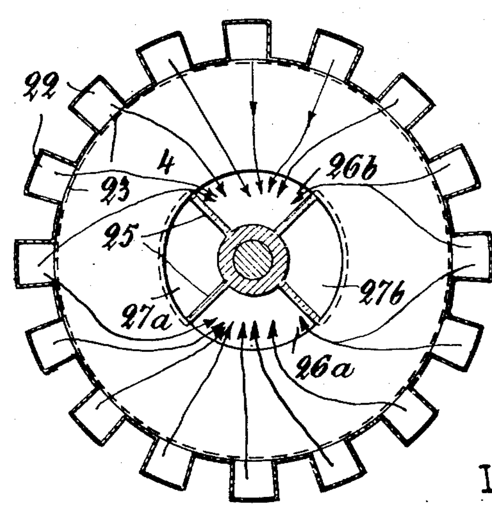

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Fig. 5.

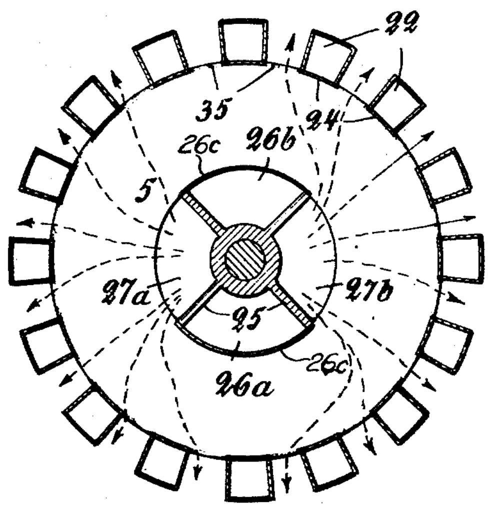
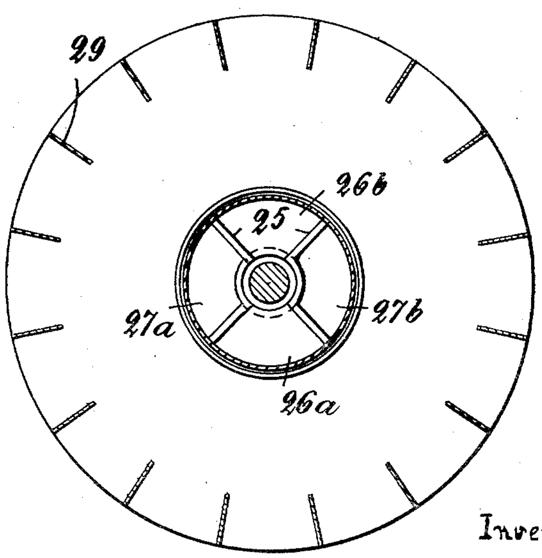


Fig. 6.



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RECUPERATIVE HEAT EXCHANGER OF THE COUNTERFLOW TYPE FOR GASEOUS MEDIA

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Fig. 7.

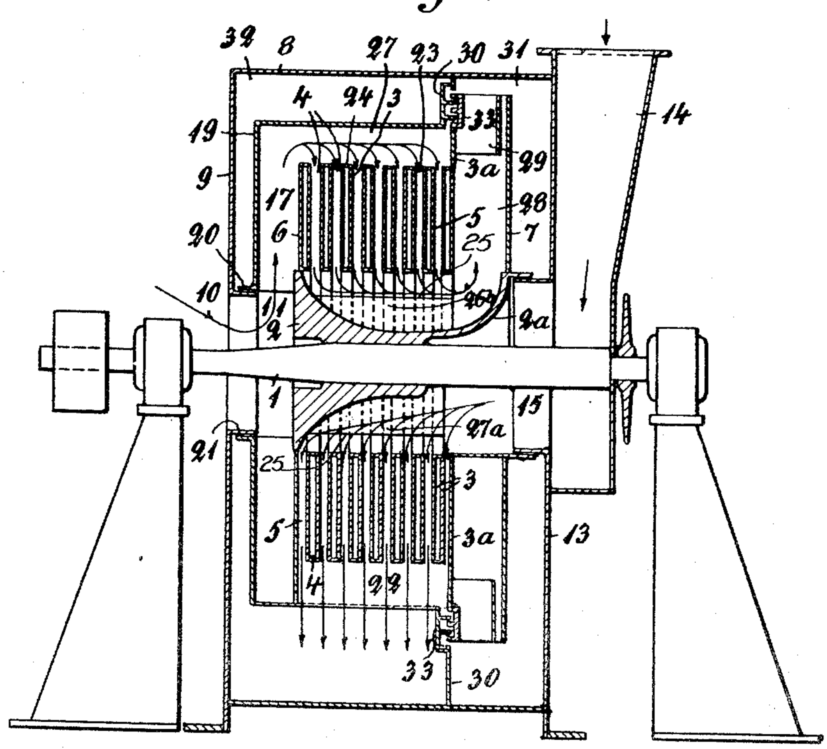

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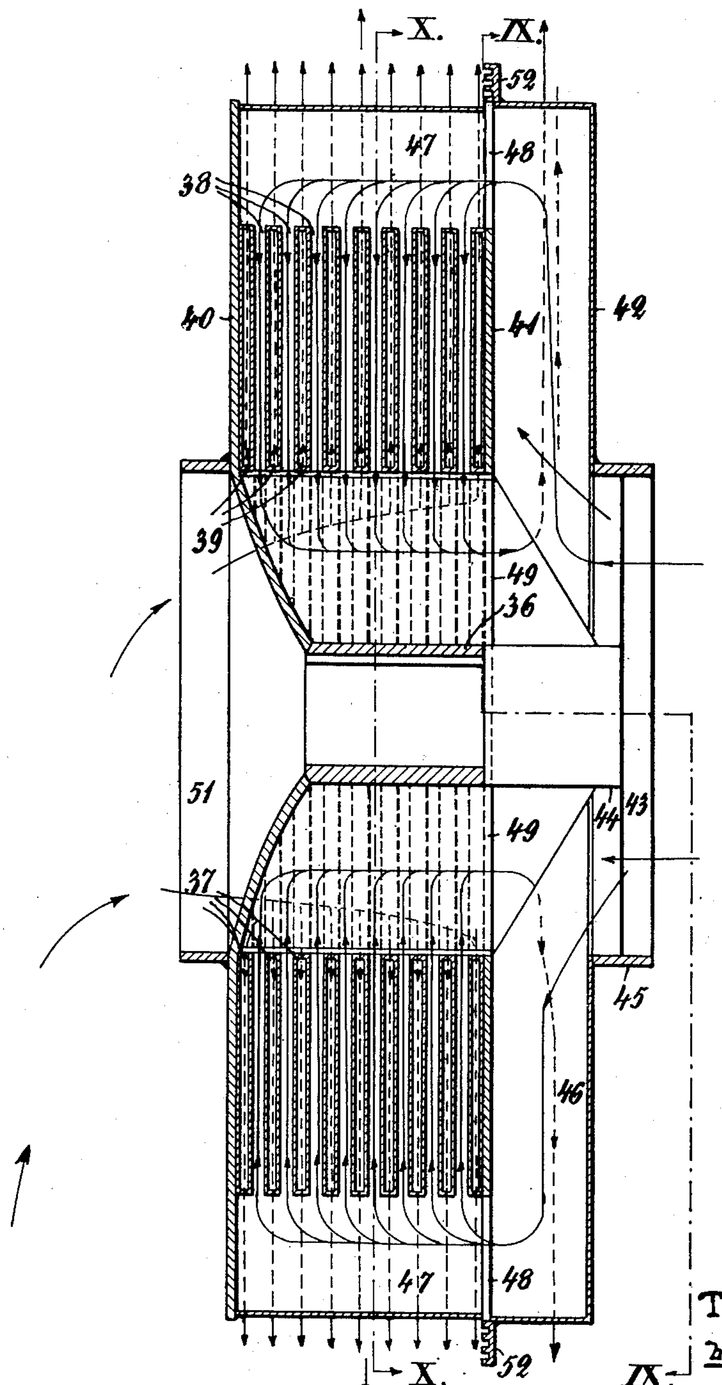
Fig. 8.



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RECUPERATIVE HEAT EXCHANGER OF THE COUNTERFLOW TYPE FOR GASEOUS MEDIA

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Fig. 9.

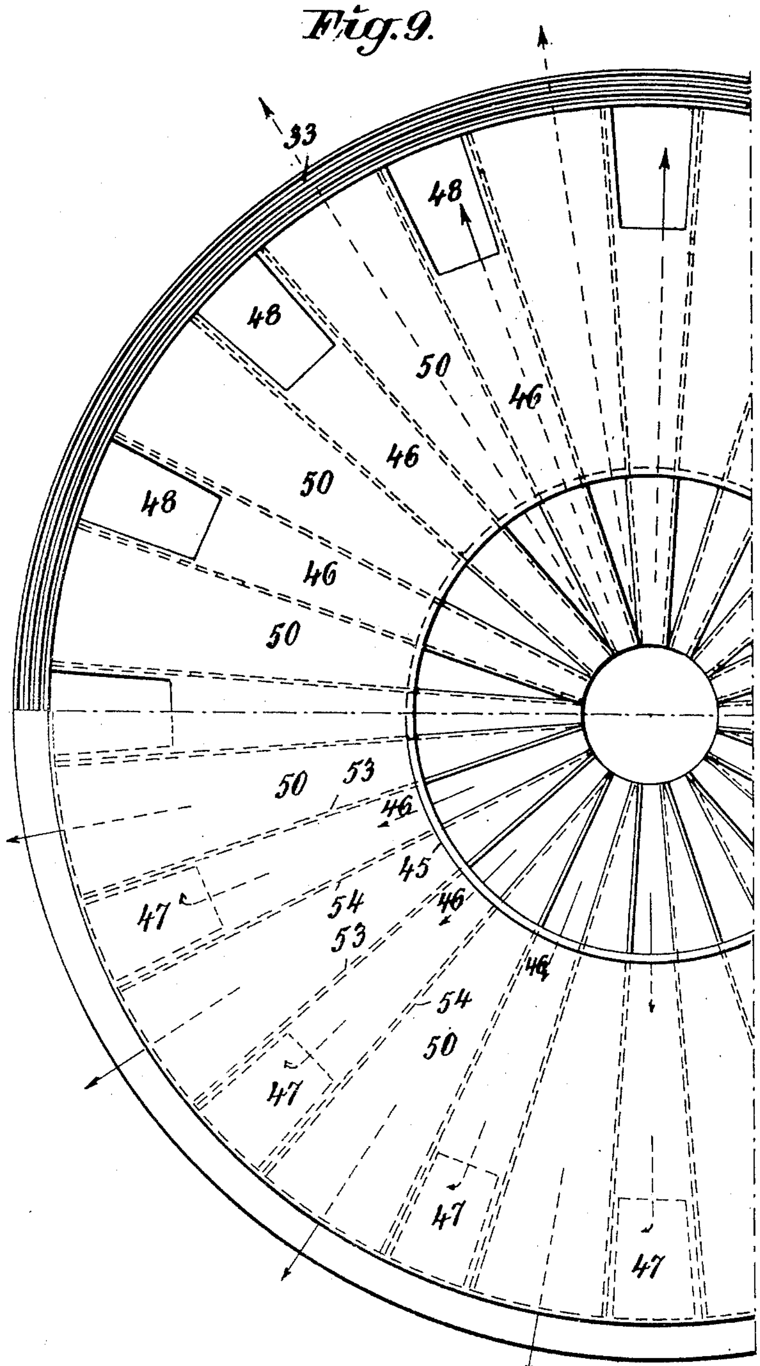

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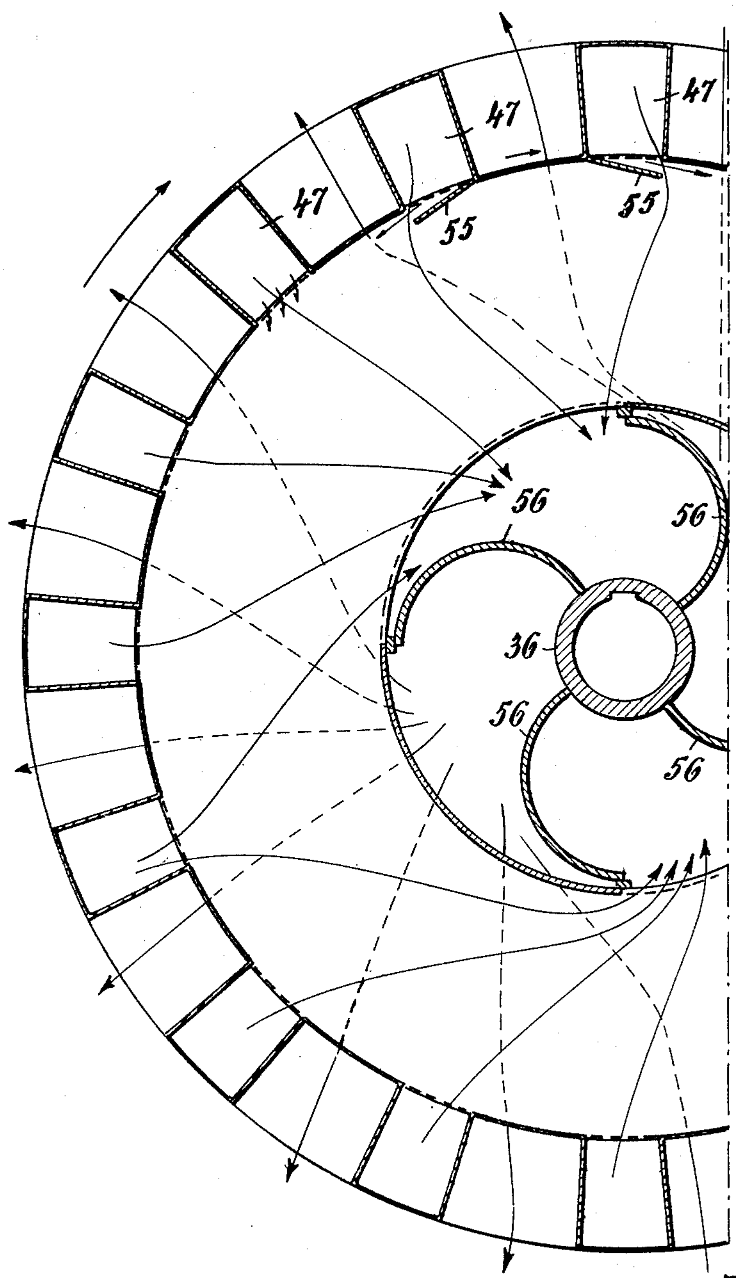
Fig. 10.



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Patented May 13, 1952 2,596,622

UNITED STATES PATENT OFFICE

2,596,622

RECUPERATIVE HEAT EXCHANGER OF THE COUNTERFLOW TYPE FOR GASEOUS MEDIA

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Application January 18, 1947, Serial No. 722,920
In Sweden September 25, 1944

Section 1, Public Law 690, August 8, 1946
Patent expires September 25, 1964

9 Claims. (Cl. 257—241)

The present invention relates to improvements in gaseous media heat exchangers of the type shown in U. S. Patent No. 2,402,307. These heat exchangers consist of a rotor arranged in a housing and fixed to a driving shaft rotor comprising a plurality of annular metal discs having substantially smooth surfaces and acting as heat exchanger members, said discs being placed substantially perpendicularly to the axis of rotation and confining a corresponding number of slot-shaped flow passages alternately for one and the other medium respectively each set of flow passages respectively communicates with a separate inlet chamber arranged adjacent to the driving shaft and with a separate outlet chamber, whereby the conveying action of the rotating discs on the gaseous media wholly or substantially in a peripheral direction is produced by the friction between the discs and the two media for the purpose of attaining at a high velocity of rotation of the discs a peripheral lagging of the gaseous media during the motion in the radial outward direction of the latter.

In the above mentioned type of heat exchangers the two heat exchanging media both pass from the inside radially outwards through the passages between the metal discs of the rotor, for which reason this heat exchanger thus operates as a direct flow apparatus. However, in many cases it is desirable that the heat exchange takes place in counter flow.

Due to the fact that the heat supplying gas by this means can be cooled down to a temperature which approaches more nearly to that of the entering cold gas it is evident that a greater heat exchange can be expected thereby.

According to the invention this is attained by having one of the two media conducted from inside and outwards through the passages of flow provided for said medium in the rotor, while the other medium by means of pressure and/or suction action is conducted from outside and inwards through the passages of flow provided for said latter medium in the rotor.

In exchanging heat between air and flue gases, the air can, for instance, be conducted from outside and inwards through the slot-shaped flow passages of the rotor, while the flue gases are conducted from inside and outwards through the flow passages of the rotor, which arrangement may be preferable from the point of view that the flue gases thus have to pass the shortest way through the heat exchanger, and that no special flue gas fan is required.

Two embodiments of a recuperative heat ex-

changer according to the invention are by way of example schematically illustrated on the accompanying drawings, in which:

Fig. 1 shows an embodiment of the heat exchanger in vertical sectional elevation, and Fig. 2 an end view of the same seen from the left-hand side in Fig. 1.

Figs. 3, 4, 5 and 6 respectively show cross-sections through the rotor along the lines III—III, IV—IV, V—V and VI—VI respectively in Fig. 1.

Fig. 7 shows the heat exchanger illustrated in Fig. 1, whereby the part above the centre line shows the passage of air through the rotor, while the part below the centre line shows the passage of gas through the rotor of the heat exchanger.

Fig. 8 illustrates in vertical sectional elevation the rotor of the heat exchanger according to the other embodiment, while Fig. 9 is a cross-section along the line IX—IX in Fig. 8.

Fig. 10 is a cross-section along the line X—X in Fig. 8 illustrating three different possibilities for one of the two media to flow into the passages of flow of the rotor from the hoods axially positioned at the circumference of the rotor.

Fig. 11 is a sectional view on the line XI—XI of Fig. 8.

In all figures the same reference numerals indicate the same parts and the arrows in solid lines and in broken lines represent the flow of air and flue gas respectively.

At the embodiment according to Figs. 1–7, **1** designates the driving shaft of the rotor. The rotor, which by means of the hub **2** is fixed to the shaft **1**, is built up of a plurality of annular, substantially smooth and preferably plane metal discs **3** set perpendicularly to the shaft, between which discs slot-shaped flow passages **4**, **5** are formed. The end discs **6**, **7** of the rotor are secured to the hub **2** in the manner shown. The rotor is enclosed by a stationary housing **8**, one end wall **9** of which is provided with an axial inlet opening **10** for one of the media, in the present case air, which opening leads to an axial inlet chamber **11** of the rotor. The housing **8**, which is spiral-shaped in relation to the rotor, is further provided with a tangential outlet **12** (Fig. 2) for the same medium.

At the opposite end of the housing, wall **13** is provided with an inlet **14** for the other medium, in the present case flue gases, which inlet opens into an axial inlet chamber **15** of the rotor. The spiral-shaped housing **8** is further provided with a tangential outlet **16** (Fig. 2) for the latter medium.

From the axial inlet **11** of the first-mentioned medium (air) proceed a plurality of radially extending channels 17, Figs. 1, 3 and 7, each of which is formed by two walls 18 arranged between the end wall 6 of the rotor and an outer wall 19. The inner edge of the wall 19 is connected to a ring 20, which cooperates with a corresponding flange 21 on the end wall 9 of the housing to form a gaseous medium seal. The outer ends of the channels 17 each communicate with an axially directed hood or channel 22 extending along the rotor immediately outside the outer edges of the metal discs 3 to the last enlarged metal disc 3 at the opposite end of the rotor. Each channel 22 communicates by openings 23 (Fig. 4) with each of the slot-shaped flow passages 4 but is separated by means of partition walls 24 from the flow passages 5. The space between the inner edges of the discs 3 and the hub 2 is, by means of four walls 25 extending radially from the hub, the elongations of which pass through the shaft, divided into four axially extending channels 26*a*, 26*b*, 27*a* and 27*b*, of which the channels 26*a* and 26*b* communicate with the flow passages 4, but are cut off from the flow passages 5 by curved walls such as 26*c*.

The channels 26*a* and 26*b* extend somewhat beyond the outermost rotor disc 3*a* and communicate with the inner inlet of a fan wheel 28 built together with the rotor, said fan wheel being provided with fan blades 29 arranged along the circumference.

The housing 8 is divided, about in the plane of rotor disc 3*a*, by a stationary partition wall 30 into two chambers 31 and 32 respectively for receiving the two different media after their exit from the rotor. The chamber 31 forms a spiral-shaped outlet channel of the fan wheel 28, 29, and the outlet channel 12 is connected to this chamber. The partition wall 30 forms a seal with the adjacent wall of the rotating fan 28 by means of a labyrinth packing 33.

In the fast rotation of the rotor the radial channels 17 bring along the medium (air) entering through the inlet 10, 11, so that said medium is exposed to the influence of the centrifugal force and is thrown outwards, whereby a certain excess of pressure arises in the channel 22. The medium flowing through the flow passages 4 from the outside and inwards is, in the shown example, further exposed to the suction action produced by the fan 28, 29 and transferred through the channels 26*a* and 26*b*, and is finally forced by this fan out into the spiral-shaped outlet channel 31 and the outlet 12.

The other medium (the flue gases), which enters through the inlets 14, 15, flows into the channels 27*a* and 27*b* through the two openings provided in the hub part 2*a* and facing these channels, and is conducted from these channels, which are separated from the channels 26*a*, 26*b*, into the flow passages 5 of the rotor (Fig. 7) and passes through these from inside and outwards, thereby exchanging heat with the other medium passing in counter flow through the flow passages 4. The passages 5 are separated at their outer circumference from the channels 22 by the partition walls 24 but are in direct communication with the chamber 32 by means of openings 35 between the channels 22 (Fig. 5), which chamber forms an eccentric outlet channel about the rotor. From the chamber 32 the medium is led out through the outlet channel 16 (Fig. 2).

The hoods or channels projecting from the circumference of the rotor will operate as fan blades which in connection with the spiral-shaped outlet channel 32 will produce a suction action on the medium flowing from the inside and outwards through the rotor.

During the flow of the gases through the smooth flow passages 4, 5 of the rotor only the friction between the discs 3 and the gases tends to bring along the latter in the direction of rotation by means of which a great lag, i. e. a high relative speed in peripheral direction, arises between the discs 3 and the passing gases, whereby the heat exchange is influenced in a favourable manner.

In the other embodiment according to Figs. 8–10, 36 designates the rotor hub which is secured to the driving shaft. Also in this embodiment of the apparatus the rotor consists of a plurality of annular, substantially smooth and preferably plane metal discs 37 arranged perpendicularly to the driving shaft, between which discs slot-shaped flow passages 38, 39 are formed. The end walls 40, 41 of the rotor are in the manner shown secured to the hub 36. The air enters through the inlet 43 arranged in an outer end wall 42 connected to the rotor itself, said inlet being confined by the central pipe 44 and the ring 45.

Between the end walls 41 and 42 radially extending channels are provided, whereby at the centre of the rotor, as shown in Fig. 9, every other channel 46 is open outwards permitting air from outside to flow into these channels. The air passes through these channels radially outwards towards the air hoods 47 extending axially on the circumference of the rotor, into which hoods the air is introduced through the openings 48.

Herefrom the air is distributed to all the passages for the flow of air through the rotor and passes through these flow passages 38 towards the centre of the rotor. From the centre portion of the rotor the air is conducted through openings 49 to the radial channels 50 provided between the end walls 41 and 42 and between above mentioned channels 46, said channels 50 being open inwards towards the centre portion of the rotor.

The air then flows through these radial channels which pass between the axially extending air hoods 47 and radially outwards into a housing (not shown) located outside the rotor. The flue gases are introduced into the rotor through the inlet 51 provided at the centre portion of the rotor and flow radially outwards through the flow passages 39 provided in the rotor for the gases and to a collecting chamber (not shown) located outside the rotor. 52 designates the seal provided between the outlets of gas and of air at the outside of the rotor. As will be seen from a comparison between the embodiment according to Figs. 1–7 and the embodiment of Fig. 8, the latter embodiment has the advantage that the total width of the rotor is considerably decreased corresponding to a reduction of at least the channel width of the radial air channels. A further advantage is that no leakage from flue gases to air or vice versa will occur at the central seals of the rotor due to the fact that the air enters and flows out on the same side of the rotor.

The air, which is introduced into the space between the pipe 44 and the ring 45, flows radially outwards through the channels 46 formed by the radially placed partition walls 53, 54 and the end walls 41 and 42, whereafter it flows through the openings 48 into the air hoods 47, and passes thereafter inwards through the flow passages 38.

From the centre portion of the rotor the air flows through the openings 49 out into the radial channels 50 located between the channels 46 to escape from the heat exchanger at the circumference of the rotor.

In Fig. 10 it will be seen that the air hoods 47 extending axially at the circumference of the rotor may be provided with throttling members 55 arranged where the air flows out from the hoods into the flow passages 38 of the rotor. As seen from the figure these throttling members may be directed in or opposite to the direction of rotation of the rotor, thus permitting the direction of flow of the air into the flow passages of the rotor determined. By means of these throttling members also the speed of the air flowing out from the hoods can be regulated. Fig. 10 also illustrates the case when no throttling members are provided, in which case the air flows radially into the flow passages will also be seen that the partition walls 56 between the air and the gas, which walls are secured to the rotor hub 36, are formed as fan blades.

The embodiment illustrated may be modified in many different ways within the scope of the invention. Thus it may be so arranged that the hot gas (the flue gases) instead is conducted from the outside and inwards through the flow passages of the rotor, while the cold gas (the air) is conducted from the inside and outwards. In the embodiments illustrated the two media are conducted to the rotor by axial inlets whereafter one of the media is conducted through radial channels out to the outer circumference of the rotor. Instead of that, one of the media (the cold or the hot one) may be conducted directly to a channel or chamber enclosing the rotor without said medium being at first axially conducted to the rotor.

Having now particularly described the nature of my invention and the manner of its operation what I claim is:

1. A recuperative heat exchanger for two gaseous media at different temperatures comprising a housing; a shaft rotatably mounted in said housing; a rotor hub secured to said shaft; a plurality of mechanically interconnected, axially spaced, substantially plane radially extending annular discs secured to rotate with said rotor hub, the spaces between said discs forming alternate gaseous media flow passages in said housing separated by intermediate flow passages; a first axial inlet in said housing for one gaseous medium; conduit means connecting said first axial inlet to the radially outer ends of alternate flow passages; first passage means at said rotor hub communicating with the radially inner ends of such alternate flow passages; a first outlet in said housing in communication with said first passage means; means sealing the intermediate flow passages from said conduit means and said first passage means; whereby such one gaseous medium will enter said first axial inlet, flow radially outward into said conduit means, then radially inward through such alternate flow passages to said first passage means, and be discharged from the latter through said first housing outlet; a second axial inlet in said housing for the other gaseous medium; second passage means at said rotor hub in communication with said second axial inlet and the radially inner ends of the intermediate flow channels; an outlet chamber in said housing in communication with the radially outer ends of the intermediate flow passages; means sealing the alternate flow passages from said chamber and said second passage means; and a second outlet in said housing in communication with said outlet chamber; whereby the other gaseous medium will flow between said discs in counter flow heat exchange relation with the first gaseous medium.

2. A recuperative heat exchanger as claimed in claim 1 including draft creating means operatively associated with said discs and having an inlet in communication with said first passage means and an outlet in communication with said first housing outlet.

3. A recuperative heat exchanger as claimed in claim 1 including a fan carried by said discs and having an inlet in communication with said first passage means and an outlet in communication with said first housing outlet.

4. A recuperative heat exchanger as claimed in claim 1 in which said conduit means comprises axially extending circumferentially spaced hoods on the peripheries of said discs.

5. A recuperative heat exchanger as claimed in claim 1 in which said first and second passage means are defined by axially extending, substantially radially projecting walls secured to rotate with said discs and said rotor hub, and said first named sealing means includes circumferentially curved wall sections interconnecting the outer ends of pairs of said walls alternately in a circumferential and in an axial direction.

6. A recuperative heat exchanger as claimed in claim 1 in which said housing outlets are tangentially directed.

7. A recuperative heat exchanger as claimed in claim 1 in which said first and second passage means are defined by axially extending, substantially radially projecting, radially curved walls secured to rotate with said discs and said rotor hub, and said first named sealing means includes circumferentially curved wall sections interconnecting the outer ends of pairs of said walls alternately in a circumferential and in an axial direction.

8. A recuperative heat exchanger as claimed in claim 1 in which said first axial inlet is annular and said conduit means comprises axially extending circumferentially spaced hoods on the peripheries of said discs, and including radial vanes forming alternate radial channels connecting each hood individually to said first axial inlet and intermediate radial channels connecting said first passage means to said first housing outlet.

9. A recuperative heat exchanger as claimed in claim 1 in which said conduit means comprises axially extending circumferentially spaced hoods on the peripheries of said discs, and including throttling means controlling flow of such one gaseous medium into the alternate flow passages.

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REFERENCES CITED

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

UNITED STATES PATENTS

Number	Name	Date
1,656,790	Heijkenskjold	Jan. 17, 1928
1,728,204	Falla	Sept. 17, 1929
2,402,307	Vannérus	June 18, 1946

FOREIGN PATENTS

Number	Country	Date
559,896	Great Britain	Mar. 9, 1944
640,183	France	Mar. 24, 1928
539,730	Germany	Dec. 1, 1931