Jan. 1, 1952 | A. BRAMLEY | 2,581,168

THROTTLING PROCESS AND DEVICE

Filed Jan. 12, 1948 | 5 Sheets-Sheet 1

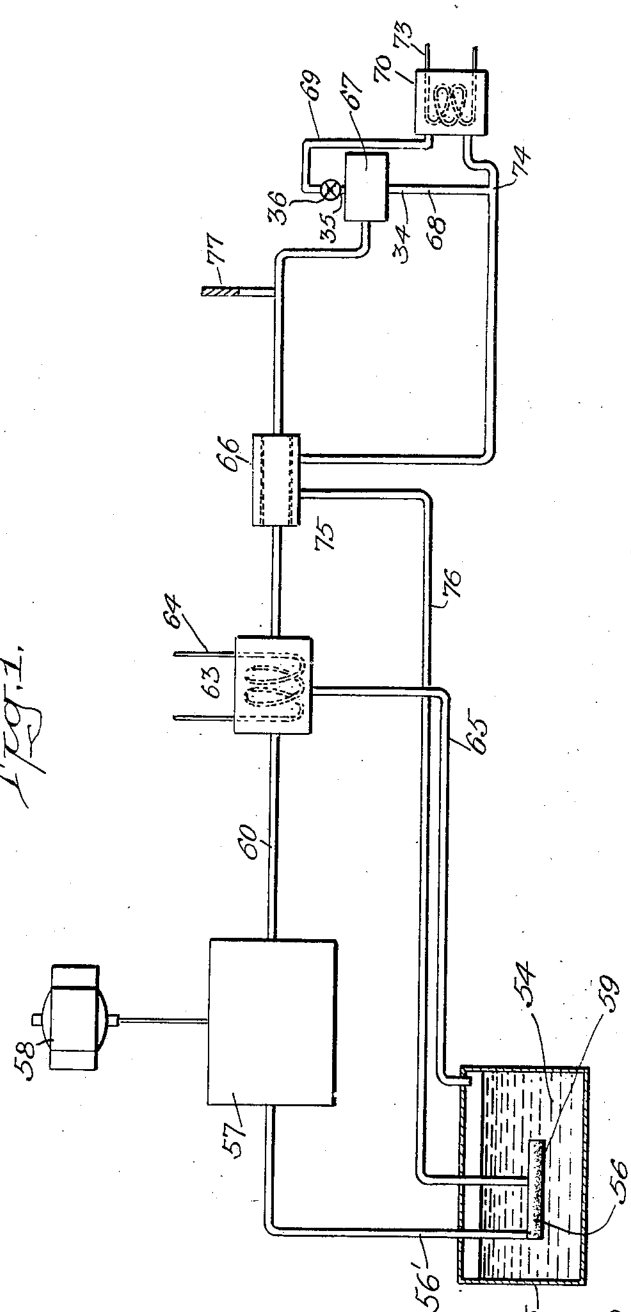

Fig. 1.

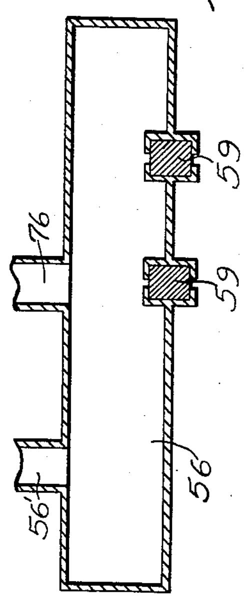

Fig. 1a.

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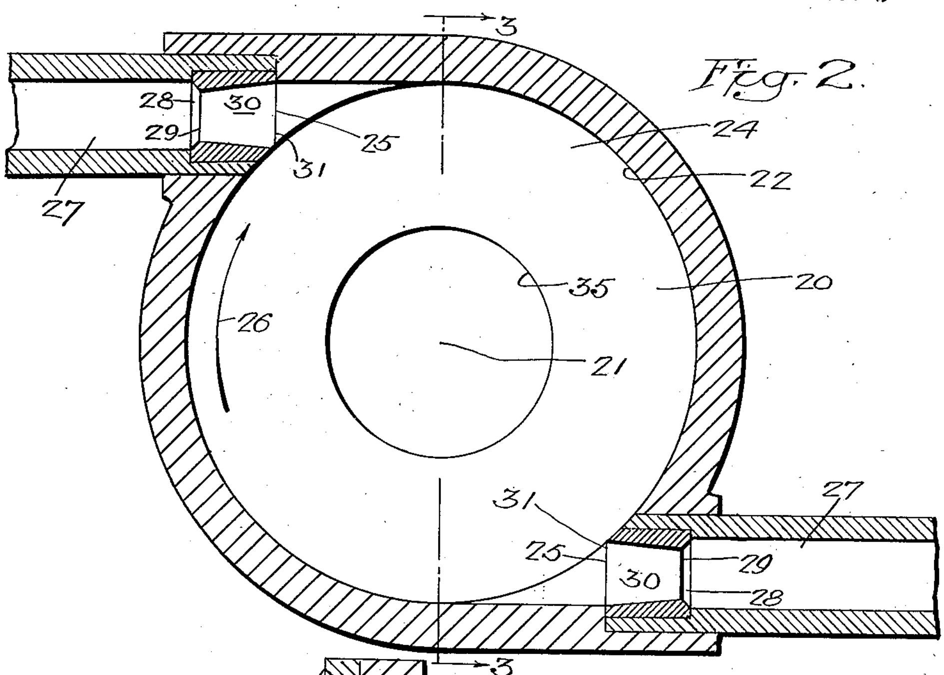

Fig. 2.

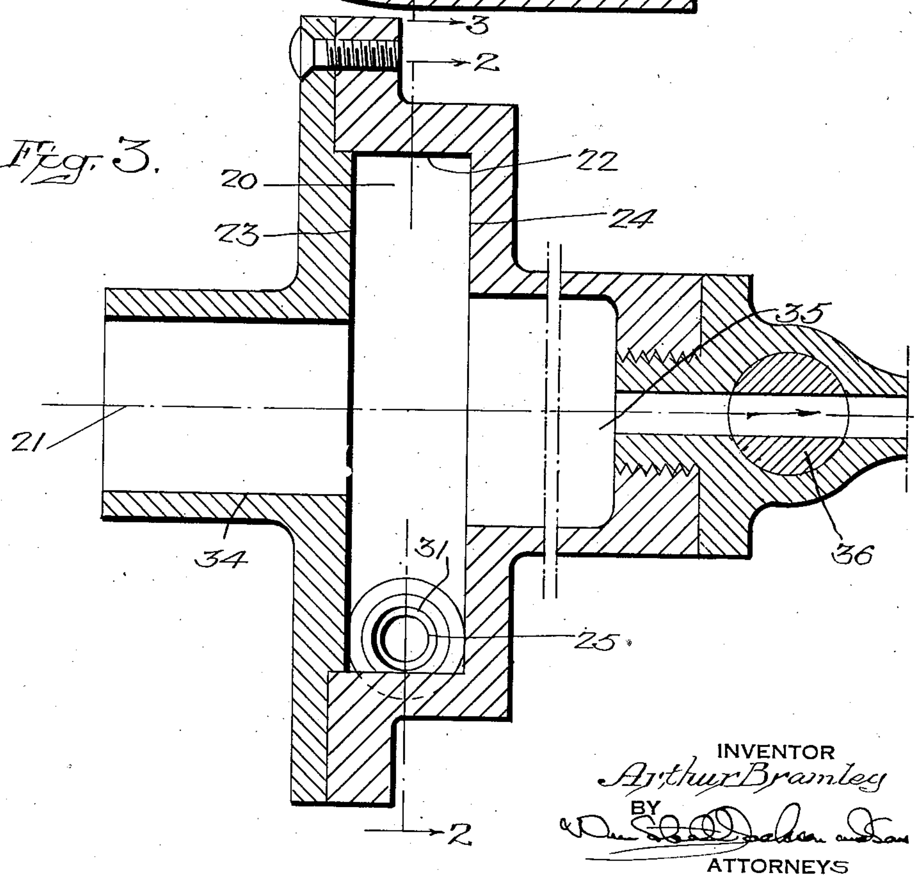

Fig. 3.

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Fig. 4.

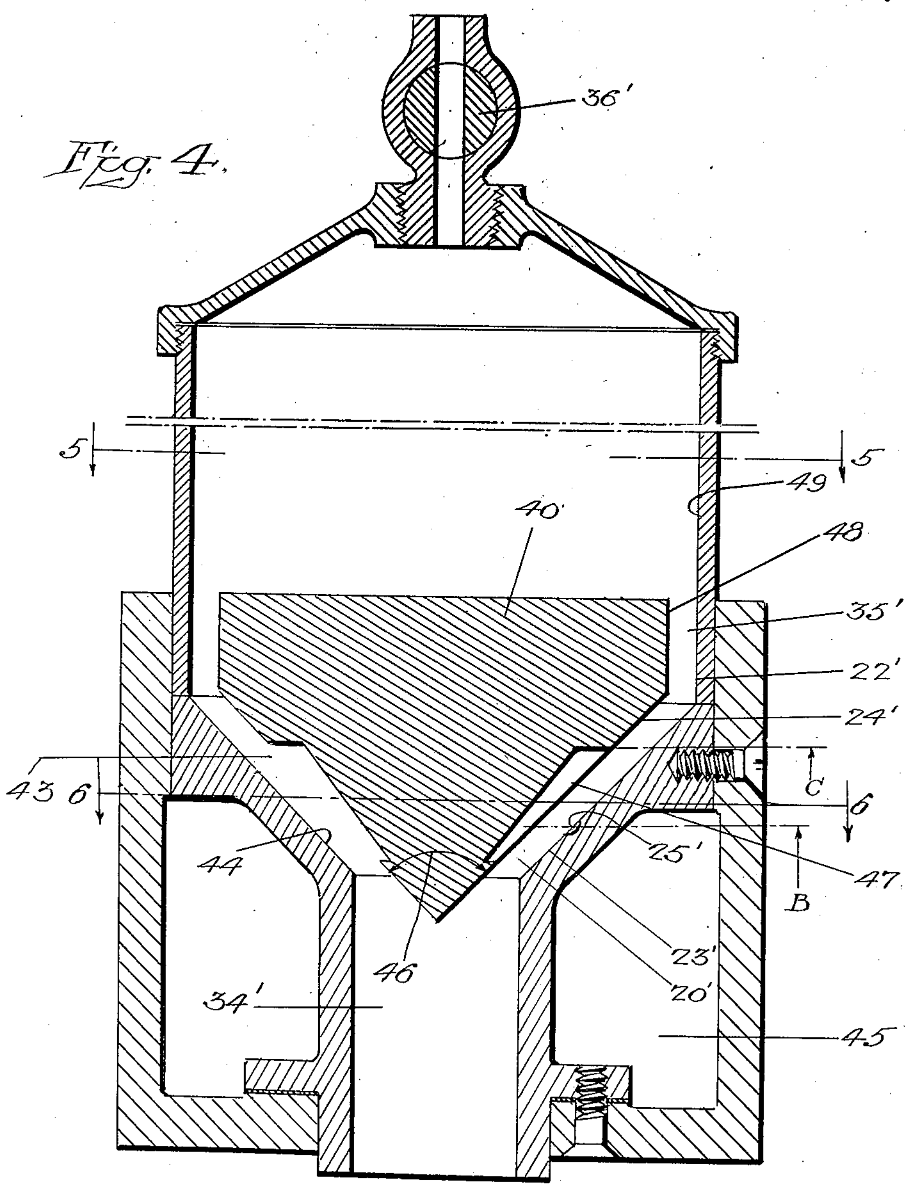

Fig. 9.

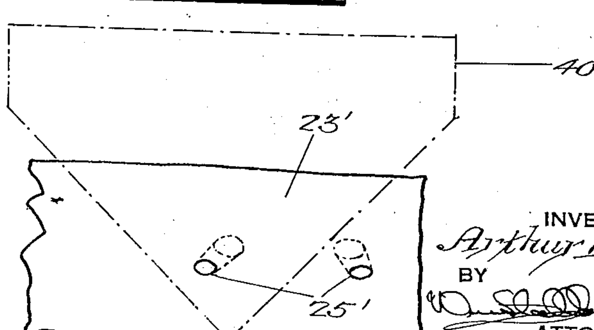

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Fig. 5.

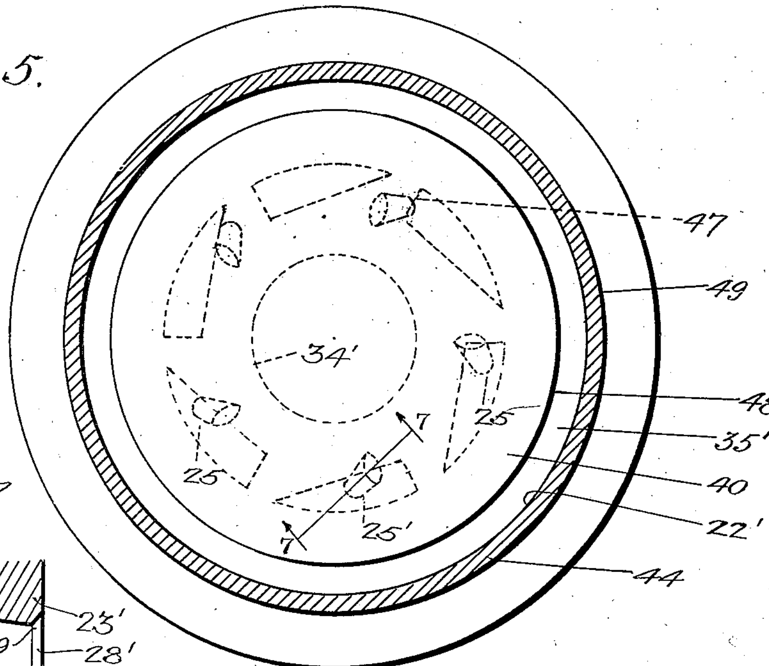


Fig. 7.

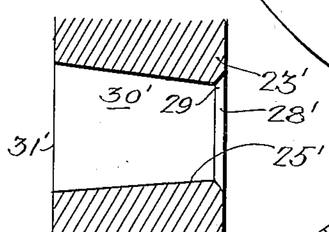
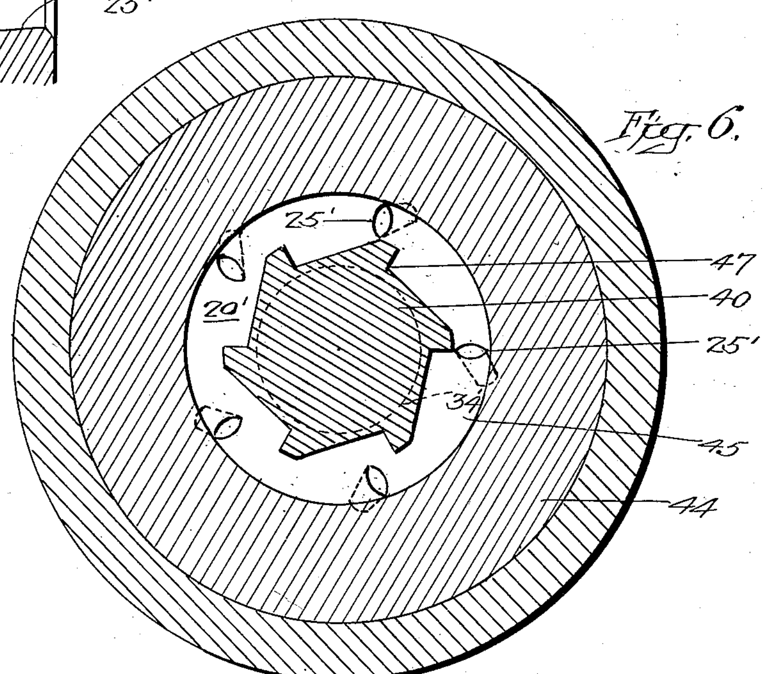


Fig. 6.



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THROTTLING PROCESS AND DEVICE

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Fig. 10.

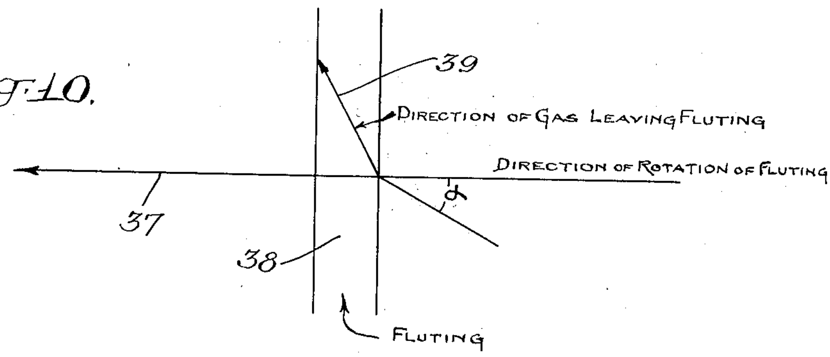


Fig. 11.

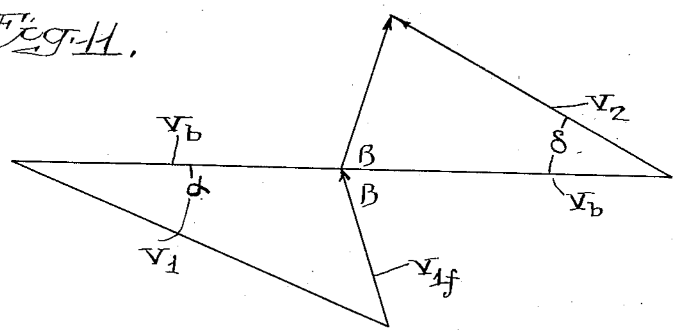
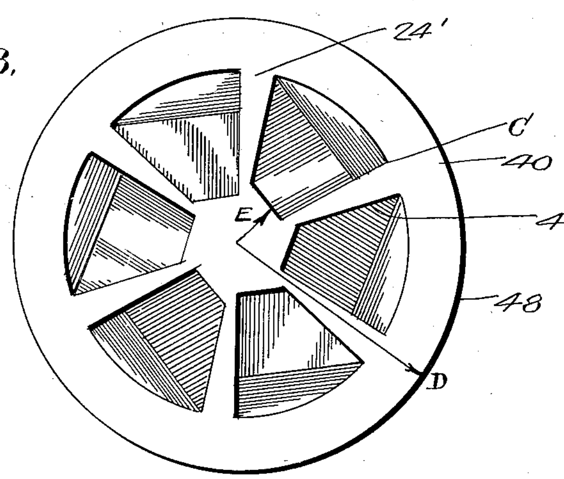


Fig. 8.



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Patented Jan. 1, 1952

2,581,168

UNITED STATES PATENT OFFICE

2,581,168

THROTTLING PROCESS AND DEVICE

Arthur Bramley, Long Branch, N. J.

Application January 12, 1948, Serial No. 1,728

11 Claims. (Cl. 62—136)

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The present invention relates to processes of throttling and of heat transfer and refrigeration thereby, and to apparatus suitable for this purpose.

A purpose of the invention is to permit the separation of compressed gas or vapor into hotter and colder fractions without the necessity of motion of the mechanism, or with motion of only a portion of the mechanism.

A further purpose is to accomplish swirling and separation into hotter and colder fractions in a swirl chamber which is partially or wholly stationary, building up a peripheral velocity in the swirl which exceeds the velocity of sound in the particular gas or vapor and accomplishing exit endwise of the swirl chamber.

A further purpose is to compress a gas or vapor, to bring a gas or vapor to an inlet under conditions of streamline flow, to cool the gas or vapor preferably by substantially adiabatic expansion in a nozzle of the inlet, with resultant acquisition of high velocity, to introduce the gas or vapor through a mouth of the inlet into a swirl chamber with the axis of the swirl generally conforming to the axis of the swirl chamber and the periphery of the swirl generally following the contour of the periphery of the chamber, to build up a velocity in the swirl suitably at the periphery which exceeds the velocity of sound for the particular gas or vapor, to preferably exert an increased pressure on the outside of the swirl compared to the pressure at the axis in excess of one-half atmosphere due to centrifugal force, to cause the hotter molecules to do work against the pressure gradient, to withdraw the hotter and colder fractions through exits endwise of the swirl chamber bearing relations as herein set forth, and to pass the hotter fraction through a constriction.

A further purpose is to provide an exit for the colder fraction through an end wall of the swirl chamber which is not removed from the mouth by a distance in excess of three times the mouth diameter, which is relatively close to the axis and extends over a relatively smaller area and to conduct the hotter fraction through an exit in an end wall, preferably the opposite end wall, at least in part further from the axis and extending over a relatively larger area.

A further purpose is to provide a valve in the exit of the hotter fraction to permit regulation of the constriction so as to adjust the relative volumes and temperatures of the two fractions and to adjust the speed of the rotor where a rotor is employed, the valve preferably being located not closer to the swirl chamber than ten times the diameter of the swirl chamber.

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A further purpose is to provide a generally cylindrical open interior swirl chamber which has straight transverse end walls in one form and cone-shaped end walls with the cones cooperating in another form, the length of the chamber being in any case small as compared to its diameter.

A further purpose is to separate condensable fluid before swirling.

A further purpose is to insulate the walls between the hot and cold exits.

A further purpose is to precool the compressed gas or vapor after compression, whether or not it is to be cooled by adiabatic expansion in the inlet means, and preferably to employ the colder fraction or a portion thereof, preferably also with the cooled hotter fraction combined therewith, for precooling.

A further purpose is to introduce the compressed gas or vapor into the swirl chamber as a jet or jets tangential to the periphery and located in the circumferential wall.

A further purpose is to maintain a diameter of the cold exhaust which is between one-half and one-third the diameter of the swirl chamber.

A further purpose is to employ a diameter of the swirl chamber which is between four and five times the diameter of the mouth of the single nozzle, or the equivalent single nozzle of the multiple nozzles, or other inlet means.

A further purpose is to utilize an inlet gas pressure on the nozzle of between 1.9 and 20 atmospheres, in many cases exceeding three atmospheres.

A further purpose is to utilize an adiabatic nozzle having an abrupt converging portion, a throat and a diverging portion having a length about five times the difference between the throat diameter and the mouth diameter.

A further purpose is to employ jets in an end wall disposed at an angle to the wall and creating a swirling component for producing the swirl and for rotating a rotor and also an axial component preferably upwardly to support or float a rotor.

A further purpose is to provide blades on the preferably cone-shaped end of a rotor forming part of the swirl chamber and to rotate the rotor by jets entering the swirl chamber and preferably supporting or floating the rotor upwardly on a vertical axis.

A further purpose is to reduce the temperature of the hot gases by causing them to do work on a rotating wall of the swirl chamber.

A further purpose is to space the rotor and stator cones by a distance transverse to the rotor cone face between one and three times the diameter of the mouth of the individual inlet means (singe nozzle where there are multiple nozzles).

A further purpose is to provide exit for the hotter fraction around the rotor endwise in an exit passage, the difference between the inner and outer diameter of which at each side is between one and three times the diameter of the mouth of the individual inlet means (a single nozzle, where there are multiple nozzles).

A further purpose is to extend the blades on the rotor vertically farther than the vertical extent of the nozzles.

A further purpose is to employ a different number of blades than nozzles.

A further purpose is to pick up gas or vapor from a heat transfer mechanism, preferably with condensable fluid from a condensable fluid bath in heat transfer relation with the heat transfer unit, to compress the same, preferably to eliminate condensable fluid from the compressed gas or vapor and preferably return the condensable fluid to the bath of condensable fluid, to cool the compressed gas or vapor, and to separate the condensed gas or vapor into fractions of different heat content as previously indicated, providing cooling of the heat transfer unit from the colder fraction and preferably also from the cooled hotter fraction, and also providing cooling of the compressed gas or vapor at least in part from the colder fraction and preferably also from the cooled hotter fraction.

A further purpose is to introduce condensable fluid to the heat transfer unit through a porous plug and permit return flow through the same means.

Further purposes appear in the specification and in the claims.

In the drawings I have chosen to illustrate a few only of the various embodiments in which my invention appears, choosing the forms shown from the standpoints of convenience in operation, satisfactory operation and clear demonstration of the principles involved.

Figure 1 is a diagrammatic view of a system to which the throttling device of the invention may be applied, the form of Figure 1 being equally applicable to either species.

Figure 1^a is a diagrammatic longitudinal section of a detail of Figure 1 showing the inlet chamber and porous plugs.

Figure 2 is a diagrammatic transverse section of the fully stationary form of my throttling device, the section being taken on the line **2—2** of Figure 3.

Figure 3 is a diagrammatic longitudinal section of Figure 2 on the line **3—3**.

Figure 4 is a diagrammatic central longitudinal section of the partially rotary form of my throttling device.

Figure 5 is a section of Figure 4 on the line **5—5**.

Figure 6 is a section of Figure 4 on the line **6—6** showing a portion of the rotor in section.

Figure 7 is a diagrammatic enlarged axial section through one of the nozzles.

Figure 8 is a bottom plan view of the rotor in Figures 4 and 5.

Figure 9 is a fragmentary elevation of the interior of the stator showing the mouths of the nozzles, with the rotor positioned on the view for placement purposes.

Figures 10 and 11 are vector diagrams which aid in explaining the invention.

Describing in illustrations but not in limitation and referring to the drawings:

The present application is a continuation in part of my copending applications Serial No. 632,825, filed December 5, 1945, for Centrifugal Dehydrating and Cooling System; and Serial No. 706,739, filed October 30, 1946, for Dehydrating, Liquefying or Cooling Gas and Air, which in turn are continuations in part of my applications Serial No. 417,960, filed November 5, 1941, for Centrifuge and Method, and Serial No. 464,509, filed November 4, 1942, for Separation of Fluids by Simultaneous Centrifugation and Selective Diffusion, Patent No. 2,422,882, granted June 24, 1947, all incorporated herein by reference. The present subject matter relates to the stationary or partially stationary swirl chamber form, whereas my application Serial No. 632,825, aforesaid, involves a fully rotary form.

The present invention relates to methods and apparatus for throttling gas or vapor to separate the use or vapor into fractions of different heat content. It will be understood wherever reference is made herein to gas that vapor is also included, as well as mixtures of gas and vapor.

It will be evident that the gas or vapor employed herein might be air, nitrogen, hydrogen, helium, argon, oxygen (using ordinary precautions), ammonia, carbon dioxide, freon, methyl chloride or other suitable medium which would remain in gaseous phase under the operating cycle chosen.

It is known that it is possible to change the heat content of a gas by cooling while at the same time reducing the pressure of the gas in a turbo-cooler, as for example that of Kapitza, U. S. Patent No. 2,280,585. Such devices involving very high speeds of turbo-rotors, with small clearances, present very serious mechanical problems of construction and maintenance, which have interfered with the general application of these machines. There are also numerous methods for changing the heat content of a gas without any appreciable change in pressure, as for example the Norst Heat Engine. See A. M. Moody, 16 J. App. Phys. 551 (1945).

The present inventor has discovered that gas revolving in a high speed swirl can be separated into hotter and colder fractions, and that providing certain conditions are met as set forth herein, the separation can be accomplished in a practical and efficient manner. This result is obtained because the hotter and faster moving molecules push away from the center of the swirl, while the colder and slower moving molecules are trapped in the swirl and eventually find their way to the center. The hotter molecules are often generating still further hotter molecules as they push away from the axis of the swirl by doing work against the pressure gradient.

The present inventor has discovered that when the peripheral velocity of the swirl exceeds the velocity of sound in the particular gas or vapor, a very unusual effect occurs, by which the colder fraction becomes markedly colder and the hotter fraction becomes markedly hotter, greatly increasing the practicability of the device and process. It has also been discovered that with increase in the temperature difference between the hotter and colder fractions a smaller volume of the gas is cooled to the temperature of the colder fraction. The present inventor has discovered, however, that machines which are most efficient for reducing the heat content of the separated gases operate at small temperature differentials.

For best results the gas or vapor which is to be separated into hotter and colder fractions should contain no condensable fluid which will produce condensable liquid. For example if the gas be air, the condensation of water vapor present in the air will reduce the efficiency of the machine to a marked extent.

For most efficient operation the portion of the swirl chamber in contact with the colder fraction and the portion in contact with the hotter fraction should be heat insulated from one another so that heat flow through the material of the swirl chamber wall will not tend to equalize the temperature differential.

It is important that the swirl chamber be circular (cylindrical) and that it have an open interior so that the swirl can follow the general contour of the periphery substantially around the axis of the swirl chamber without formation of eddies which will dissipate kinetic energy as heat. The inlet should be so designed that a peripheral velocity in excess of the velocity of sound can be built up in the swirl as later explained. While other suitable inlets may be employed, the preferred inlet is through a substantially adiabatic nozzle, having first a region of sharp convergence, then a minimum or throat and finally a region of more gradual divergence, preferably having a length about five times the difference between the diameter of the mouth and the diameter of the throat through which the nozzle discharges (see Everett, Thermodynamics, chapter VIII and Stodola, Steam and Gas Turbines, Volume I, section 19, and chapter X, section 167). The critical pressure ratio above which the nozzle must function is 1.90 for air.

If the gas or vapor is introduced into the swirl chamber at high speed but at moderate temperature, such as room temperature, the temperature of the colder fraction will be markedly higher than if the gas or vapor is cooled before introduction. The cycle may be carried out in various ways. The initial cooling of the gas or vapor after compression and before entry into the swirl chamber at high speed may be accomplished by a precooler using as a coolant a part of the colder fraction from the swirl chamber, or may be accomplished by an adiabatic nozzle in the inlet to the swirl chamber or by a turbocooler or any combination of these. By this procedure a continuous lowering of the initial temperature of the compressed gas takes place, which is very desirable where the colder fraction is to have the lowest possible temperature.

In the preferred embodiment, condensable fluid such as water vapor will be removed from the gas or vapor before it enters the throttling device. Conventional devices such as water eliminators may be used for this purpose.

The swirl space is circular (cylindrical) and has an open interior so as to permit swirling unimpeded. The end walls of the chamber in which the gas or vapor swirls may be straight and transverse or may be frustums of a cone. Where the straight transverse ends are used the inlet will preferably be tangential through the periphery, but where the conical ends are employed the inlets are preferably placed not at the periphery but in an end wall at a distance from the periphery, and are oriented so as to

produce a radial component outwardly and a component in a direction parallel to the axis and away from the end wall through which the inlet is accomplished.

The conical form will preferably have its axis vertical and will be provided with a rotor in one (desirably the upper) end wall which will rotate due to the swirl and preferably be supported on the jet or jets. The end wall rotor is suitably provided with blades or flutings on the conical surfaces which are oriented with respect to the direction of the jet or jets so that the gas or vapor does work on the rotor lowering its heat content.

In the various forms a constriction is provided in the exit for the hotter fraction, preferably as a valve which can be adjusted to determine the temperature difference and volume relation between the hotter and colder fractions. Where the rotor is used the valve will also regulate the speed of rotation.

The swirl chamber will have a length which is short compared to the diameter, the critical feature being that the inlet mouth is quite close to the end wall through which the exit for the colder fraction leaves the chamber. The mouth distance from the end wall through which the colder fraction leaves the chamber will not exceed three times the mouth diameter, and in many cases the mouth will be directly in line with the end wall at one side, or actually in that end wall. The mouth diameter here referred to is the diameter of the individual mouth (one of several, where multiple nozzles are used).

Only in the line of the nozzle mouth is the velocity a maximum. As one moves away from that region in the direction parallel to the axis near the periphery, the velocity of the stream falls. In such region of lower pressure along the periphery, the gas becomes hotter due to diffusion of hotter molecules out of the high speed gas stream. In the direction of the hot exhaust, the hotter molecules are drained off, but in the direction of the cold exhaust the hotter molecules would diffuse into the cold stream and raise its temperature, defeating the purpose of the device, if space were left for them to accumulate. The limitation of the distance between the mouth and the end which contains the cold exhaust serves to limit the entry of hotter molecules into the colder fraction.

The swirl will build up a pressure in the outside due to centrifugal force which will exceed the pressure at the axis of the swirl by at least one-half atmosphere and usually by three-quarters of an atmosphere. This is due to the fact that the velocity of the swirl exceeds the velocity of sound, and to the fact that at room temperature the molecular weight of the gas under consideration is less than or equal to that of air. The pressure ρ at the periphery of the swirl is related to the pressure ρ_0 at the axis as follows:

$$\rho=\rho_0 \exp \frac{M\omega^2\Gamma^2}{2RT}$$

where M is the molecular weight of the gas, R is the gas constant, T is the temperature, Γ is the radius of the swirl chamber,

$$\frac{\omega}{2\pi}$$

is the frequency in revolutions per second, and $\omega\gamma$ is the peripheral velocity. Hotter molecules must do work against the pressure gradient in the swirl.

The exits from the swirl chamber will extend endwise and preferably parallel to the axis, but will have different areas and extend to different distances from the diameter. The exit for the colder fraction will also be in a wall which is close to the mouth as above indicated. Of the two exits, the exit for the colder fraction will be closer to the axis and the exit for the hotter fraction will extend at least in part farther from the axis. Of the two exits, the exit for the colder fraction will have the smaller cross sectional area as it leaves the swirl chamber. The two exits will be preferably, but not necessarily, coaxial with the chamber.

The preferred diameter of the exit for the colder fraction will be between one-half and one-third the diameter of the swirl chamber, as it has been found that turbulence is minimized in this way. The preferred diameter of the swirl chamber is between four and five times the diameter of the inlet mouth, in order to obtain most efficient results. In this case the diameter of the mouth where multiple nozzles are used is the diameter of the mouth of the single nozzle which will pass the same amount of gas at the same speed as the multiple nozzles (called the single equivalent nozzle). In the case of a single nozzle, the single nozzle and the single equivalent nozzle are the same.

Considering first the throttling device of Figures 2 and 3, the swirl chamber **20** is cylindrical, having an axis **21**, a circumferential peripheral wall **22**, a straight transverse cold end wall **23**, and a straight transverse hot end wall **24**. The interior of the swirl chamber is entirely open and unimpeded by vanes, partitions, walls or structure of any other character.

Inlet to the swirl chamber is accomplished through the inlet means **25**, here shown as a nozzle. It will be understood that any suitable number of nozzles may be used, the drawing being limited to the illustration of two nozzles, rather than to one, three, four or some other number in order to simplify the illustration, and multiple nozzles being shown in Figures 4 to 9, intended to indicate that they may be used in any of the forms.

The preferred embodiment has the nozzle tangentially directed to the swirl chamber as shown so that it will impinge upon the periphery of the swirl **26** which rotates in the swirl chamber on the axis **21**. Each nozzle **25** has an approach portion **27** through which streamline flow is preferably obtained, a converging portion **28**, a throat **29**, a diverging portion **30** and a mouth **31**. The nozzle is preferably an adiabatic expansion nozzle, in which case the diverging portion will preferably have a length of approximately five times the difference between the mouth and the throat diameters.

The adjoining edge of the mouth is distant from the cold end wall **23** not in excess of three times the mouth diameter (the actual, not the equivalent mouth diameter). In the present case the spacing is only a fraction of the mouth diameter.

The diameter of the swirl chamber is preferably between four and five times the mouth diameter. In this case, where several nozzles are used, the mouth diameter is that for the equivalent single nozzle.

Located relatively close to the axis, having a relatively smaller diameter, and extending through the cold wall endwise, is an exit **34** for the colder fraction. Arranged in one of the end walls and preferably in the opposite end wall, and extending endwise and preferably axially, is an exit **35** for the hotter fraction, which extends at least in part to a greater diameter or distance from the axis than the exit for the colder fraction and which has a larger area. When it is stated that the exit for the hotter fraction extends at least in part to a greater diameter, it is intended to indicate that the portion of the exit **35** near the axis may be optionally open or closed, but some portion of the exit for the hotter fraction will extend farther from the axis than the exit for the colder fraction, and the over-all area of the exit for the hotter fraction as it leaves the swirl chamber will be greater.

The coaxial arrangement with respect to the exits and with respect to the chamber is preferred.

The relations given for the dimensions of the mouth, the swirl chamber and the exits assure efficiency great enough to make the device practical and the preferred ranges as above specified insure that maximum efficiency will result.

The material and thickness for the swirl chamber walls should be chosen so that heat conductivity will be reduced to a minimum. Thus the walls are to advantage made of plastic, such as phenol formaldehyde or acrylic plastic, glass or ceramic, where the pressures will permit, or are conveniently made of metals such as stainless steel, Nichrome, or inconel having lower heat conductivity, or of metals having enameled, ceramic, plastic or rubber coatings to reduce heat conductivity.

The annular peripheral wall **22** of the swirl chamber may desirably have a thickness sufficient so that all or part of the inlet can be constructed in the wall, as for example by making the entire nozzle or nozzles as ports in the wall, or by forming a portion of the nozzles as such ports.

In the form shown, the nozzles are constructed as inserts in the peripheral wall **22**. The length of the channel as compared with its diameter and the shape of the channel depend upon the particular design used. In the preferred form for adiabatic nozzles the length of the portion **30** is about five times the difference between the mouth and the throat diameters. In order to permit the adiabatic nozzle to operate above the critical pressure ratio between the initial pressure and the throat pressure of 1.90, the channel **30** from the throat to the mouth should preferably diverge at an angle of about ten degrees, with a very short rapidly converging portion **28** from the nozzle inlet to the throat, which should preferably have a length of about one-twentieth of the length of the diverging portion.

The passage **27** which supplies the gas or vapor to the nozzle or nozzles should preferably have a diameter large enough compared to the diameter of the mouth so that gas or vapor flows in the passage **27** at a relatively slow rate, less than the velocity of sound in the particular gas or vapor, and preferably less than about a thousand feet per minute in air, so that it does not tend to heat up by friction effects. The approach passage **27** will therefore have streamline flow. The limiting value of the velocity v, for which streamline flow is possible, is given by the expression:

$$\frac{vd\rho}{\mu}=2,000$$

where ρ is the density, d the diameter of the approach passage **27** and μ is the absolute viscosity. See Marks Engineering Handbook (4th edition, 1941), 265. Frictional heat should be avoided at this stage as it counteracts the cooling effect and represents waste energy.

It is very important that the exit speed at the mouth and the peripheral velocity of the swirl exceed the velocity of sound in the particular gas or vapor. Thus in air the exit speed and peripheral velocity will exceed ten hundred and eighty feet per second, the acoustical velocity. To obtain this acoustical velocity it is best to employ the adiabatic type nozzle already described, so that the ratio of the initial pressure to the throat pressure exceeds 1.90 (Everett, Thermodynamics, chapter VII). Thus the gas or vapor is preferably substantially adiabatically expanded in the nozzle with resultant cooling and acquisition of high velocity.

The feature of reducing the temperature of the gas and increasing the kinetic energy can be accomplished by devices other than the typical adiabatic nozzle referred to, such as the De Laval nozzle.

The swirl in the swirl chamber **20** will exert centrifugal force when the periphery is moving above the acoustical velocity, so that the outer swirl is subjected to an increased pressure as compared to the axis of the swirl which will be in excess of one-half atmosphere and preferably of the order of three quarters of an atmosphere (these are practical values for separation into fractions of different heat contents as explained herein).

During the operation of the swirl, the faster molecules are compelled to do work against this pressure gradient, thus lowering their heat content.

The exit **35** of the hotter fraction has a constriction **36**, preferably in the form of a valve to permit adjustment. For most efficient operation the constriction should be located at a distance from the swirl chamber not less than ten times the diameter of the swirl chamber in order to reduce turbulence of the hotter fraction, and obtain the best results in adjusting the volumes of the fractions and the temperatures relative to one another.

The gas or vapor through the passage **27** will normally be at a substantial pressure, ordinarily from 1.9 to 20 atmospheres, and usually greater than three atmospheres. Higher pressures may be used. The exit of the hotter fraction at **35** will normally be at a low pressure, usually not much above atmospheric pressure, but slightly higher than the colder exhaust. The minimum inlet pressure for effective operation of the device will vary depending upon how many nozzles are used and the character of nozzle, but it should be sufficient to maintain a peripheral velocity of swirl in excess of the acoustic velocity and to maintain a difference in pressure between the circumference and axis of the swirl in excess of one-half atmosphere and preferably of the order of three quarters atmosphere or greater. The colder fraction leaves its exit **34** at a pressure which will be atmospheric in many applications. Of course, it will be evident that if the device is working on a closed cycle, the colder and hotter fractions may be exhausted at higher or lower pressures provided there be a sufficient pressure differential between the pressure of the colder and hotter fractions and the pressure in the approach passage **27**.

In case the inlet gas or vapor contains an appreciable amount of condensable vapor such as water vapor, which is not eliminated before the gas or vapor reaches the inlet mouth, the temperature drop for adiabatic flow in the diverging portion of the adiabatic nozzle will be decreased considerably since the heat of condensation is in general large as compared to the specific heat in various gases or vapors.

In many applications the hotter exhaust will be at a comparatively high temperature, and heat insulation is therefore rather important in such cases. Where the pressures are such that the walls of the swirl chamber and the adjoining hotter and colder exhausts must be made of metal, it is preferable to avoid metals of high conductivity such as copper and aluminum and to employ metals of lower heat conductivity such as stainless steel, Nichrome or inconel. For many installations heat conductivity can be further reduced without departing from metal as a construction material by employing ceramic, enamel or other coatings of lower heat conductivity on the walls of the interior of the swirl chamber and the hotter and colder exhausts. For installations where the pressure is low enough, or for portions of the walls which are subjected to lower pressures, it is satisfactory and from the standpoint of heat conductivity it is very advantageous to construct the device of plastic such as phenol formaldehyde, urea formaldehyde, or acrylic resin, or of glass or ceramic.

For many types of installations it has been found satisfactory to make the diameter of the swirl chamber of the order of three-quarters of an inch. The diameter of the swirl chamber ordinarily will not greatly exceed one and one-half inches. Successful results are obtained with swirl chamber diameters as small as one-half and one-third inches. The nozzle mouth diameter has in some cases been one-sixteenth of an inch and in individual installations has varied from one-tenth to one-thirty-second of an inch. The peripheral wall of the swirl chamber has in individual cases been one-eighth to one-quarter of an inch thick. The number of nozzles equally spaced tangentially around the swirl chamber in the preferred embodiments may vary from one to six. The flow in the exits has preferably been in the order of about one hundred feet per minute, as contrasted with a peripheral velocity in the swirl in excess of the acoustical velocity.

In an individual test on a device of the character shown the following data were taken:

Room temperature, 17° C.
Pressure of inlet air, 100 p. s. i.
Temperature of inlet air at ambient.
Volume of free air flow, ten cubic feet per minute.
Temperature of cold fraction, −4° C.
Temperature of hot fraction, 38° C.
Volume of free air cold fraction, 5 cubic feet per minute.
Volume of free air hot fraction, 5 cubic feet per minute.

It is possible to modify the shape of the end walls of the swirl chamber from the straight transverse construction shown in Figures 2 and 3. This may also include change in the position of the inlet jets and change in the relationship between the temperature and quantity of gas or vapor in the two fractions. For example, the end walls of the generally circular or cylindrical swirl space may be conical or conoidal as shown in Figures 4 to 9, with or without the feature of rotation of a portion of the end wall as there shown.

As shown in Figures 4 to 9, inclusive, a part of the wall of the swirl chamber may move. Thus the temperature of the hotter fraction can be reduced if the end wall of the chamber through which the hot gas or vapor is exhausted is designed so that the hot stream of gas or vapor can do work upon the end wall. When such end wall is made to rotate while engaged by the gas or vapor jets, the whole mass of gas or vapor assumes a lower average temperature.

As shown in Figures 10, if the direction of rotation of the fluting is **37**, the blade or fluting on which the work is done by the gas is **38**, and the direction of the gas leaving the fluting is **39**. Figure 11 represents a velocity vector diagram showing the work done by the gas or vapor on the fluting. V_1 represents the velocity of the gas jet striking the fluting at an angle α with respect to the direction of rotation **37**. V_b is the velocity of the fluting. V_{1f} is the velocity of the jet of gas or vapor striking the fluting expressed with reference to the fluting. The angle β represents the true angle at which the stream of gas or vapor enters the fluting. If the gas or vapor encounters no friction, it leaves the fluting at the same speed V_{1f} and at the same angle β relative to the fluting. The absolute velocity of the gas leaving the fluting is obtained by laying out again V_b and closing the triangle. The absolute velocity of the gas leaving the fluting is V_2 and it makes an angle δ with respect to the direction of rotation of the fluting. The force F produced by changing the direction of motion of the gas jet involving W pounds of gas per second, where g is the acceleration of gravity, is:

$$F=\frac{W(V_1 \cos \alpha+V_2 \cos \delta)}{g}$$

The work done through the distance V_b which the fluting travels per second is therefore:

$$\text{Work}=\frac{WV_b(V_1 \cos \alpha+V_2 \cos \delta)}{g}$$

This is work taken from the kinetic energy of the gas stream.

The design may be varied in respect to the shape of the end walls, the orientation of the jets with respect to the end walls and the design of the flutings or blades which act as the blades in a gas turbine.

The device of Figures 4 to 9 functions in general in the same manner as the device of Figures 2 and 3. The rotor **40** rotates on a gas cushion **43** in the swirl chamber **20'** provided between the rotor **40** and the stator cone **44**. The periphery of the generally circular swirl chamber is circular at **22'**, but the cold end wall **23'** and the hot end wall **24'** are conical and preferably conform approximately to the same cone.

Gas or vapor compressed suitably to a pressure preferably between 1.9 and 20 atmospheres or higher, and usually above three atmospheres and preferably cooled to room temperature is held in an annular reservoir **45** in the stator. From the reservoir the compressed gas flows through inlet means **25'**, preferably one of a series of adiabatic nozzles, as already described. In flowing through the nozzles located in the cold wall **23'** in this case, the high pressure gas lowers its temperature and increases its velocity to a velocity at the mouth which exceeds that of sound.

The distance between the end wall **23'** of the

stator and the end wall of the rotor measured transversely to the cone face on the rotor should not exceed three times the diameter of the individual nozzle mouth (not the equivalent nozzle) and should not be less than one times the diameter of the mouth, so that the jets of gas from the nozzle do not become excessively diffuse before impinging on the radial flutings of the rotor. Since the rotor is floating this can be adjusted by the weight of the rotor in relation to the pressure.

In some cases it may be desirable to make the cone angle **46** of the rotor slightly different from the cone angle of the stator to compensate for inequalities in air flow, but in the form shown the two cone angles have been made the same.

As best seen in Figures 4, 6 and 8 the end wall **24'** of the rotor is fluted with blades or channels **47** distributed radially around the apex of the rotor. The flutings or blades terminate at a distance E from the apex which is small as compared to the diameter D. The flutings extend to a height C above the apex which is greater than the height B of the nozzle exits from the same. In the preferred embodiment, the flutings extend over the bulk of the radial extent of the end **24'**, and extend generally in a radial direction along such end wall.

With the particular axial length of the flutings and the speed of rotation, the gas or vapor striking the flutings should do an amount of work which is appreciable as compared with the kinetic energy of the gas or vapor. If the number of nozzles is n, the number of flutings should be $nm\pm1$, where m is an interger, such as 1, 2, 3, etc.

The hot fraction of the gas escapes through the exhaust exit **35'**, provided in the clearance between the outside circumference **48** of the rotor and the tubular wall **49** of the exit. This clearance should conform to the dimensions specified for the hot gas exit in the previous discussion, but it has been found that a clearance of a few tenths of an inch (ordinarily not over one-half inch) is ample in most installations. This clearance all around the rotor can to advantage in many applications vary between one and three times the diameter of the mouth of the individual nozzle (not the equivalent nozzle). A constriction **36'** in the form of a valve is placed a distance above the rotor which exceeds ten times the diameter of the swirl chamber in order to reduce turbulence and friction. Adjustment of the valve permits adjustment of the relative volumes and temperature of the hotter and colder fractions and of the speed of the rotor.

The cold fraction is exhausted through the cold exit **34'** which joins the stator cone adjacent to the apex thereof and extends in an axial direction.

The orientation of the inlet means **25'** is such that the gas stream leaving the jets does a maximum of work on the rotor. For the flutes shown in Figures 4 to 9, the nozzles **25** should desirably make an angle of about 45° with the vertical as shown in Figure 9, assuming that the angle at the apex of the rotor is 90° as shown. The jets are arranged with respect to the flutings of the rotor so that the resultant effect is a downward acceleration on the rotor which opposes the upward force resulting from the deceleration of the gas stream on the lower surface of the rotor. The rotor is pushed down toward the stator cone because the atmospheric pressure (or whatever the exhaust pressure may be) on its top is greater than the average pressure on the lower surface. At the axis of the rotor the pressure is close to atmospheric (or to the particular exhaust pressure) whereas at the region of the maximum of the swirl where the gas is moving with high velocity after just leaving the nozzles, the pressure is low. According to Bernoulli's theorem, the pressure P and the velocity V are connected by the following equations:

$$\frac{P}{y}+\frac{V^2}{2g}=\text{constant}$$

where y is the specific weight of the gas and g is the acceleration due to gravity. Where the velocity V is negligible, as for example at the axis, because the nozzles are directed outward, the pressure P approaches the atmospheric pressure (or the particular exhaust pressure), whereas, where V is of the order of the acoustical velocity, the pressure falls a fraction of an atmosphere.

Good results may be obtained with variation of the orientation and spacings of the jets and flutings from those described above.

In the partial rotor form, because of the high speeds involved, the rotor should preferably be of metal (such as stainless steel, Nichrome or inconel) but the other parts of the equipment may be of the materials previously mentioned.

In the particular device in a specific embodiment, the rotor is about one inch in diameter and has a clearance from the stator at the circumference of about one-eighth inch all around. In operation the following data were obtained:

Number of nozzles, 7.
Number of blades, 8.
Pressure of the air in the reservoir, 50 p. s. i.
Room temperature, 81° F.
Temperature of the compressed air at ambient.
Temperature of the cold fraction, 64° F.
Temperature of the hot fraction, 86° F.
Speed of the rotor between 500 and 1,000 revolutions per second.

In this form, the rotor floated on gas as it turned.

In both of the forms of the invention as described, the inlet gas or vapor is compressed to a reasonably high pressure, preferably in excess of about three atmospheres, and is carried to the point of inlet to the throttling device preferably under streamlined flow. Inlet is accomplished through a mouth which is preferably the mouth of a nozzle suitably of adiabatic type. In the nozzle the gas is cooled and increased in velocity so that at the mouth of the nozzle and in the periphery of the resulting swirl, the velocity exceeds the velocity of sound in the particular gas or vapor. If a nozzle is not used, and preferably where a nozzle is used, the gas or vapor will be cooled before it passes through the nozzle.

The swirl chamber is open in interior and of circular or cylindrical shape so that the swirl forms about the axis of the chamber. The pressure in the periphery of the swirl of the stationary form is increased by the centrifugal force in excess of about one-half atmosphere and the faster molecules are caused to do work against the pressure differential. The mouth of the nozzle is distant from the end walls through which the cold exhaust takes place not in excess of three times the diameter of the mouth of the individual nozzle. The cold exhaust takes place closer to the axis and over a relatively smaller area than the hot exhaust which takes place at least in part further from the axis and over a relatively greater area. Flow in the hot exhaust is constricted.

In the preferred embodiment the diameter of the cold exhaust is preferably between one-third and one-half the diameter of the swirl chamber to minimize turbulence. The diameter of the swirl chamber is preferably between 4 and 5 times the diameter of the equivalent nozzle mouth (the nozzle which passes the same amount of gas or vapor at the same speed as the total of the multiple nozzles).

It should be emphasized that the throttling device of the invention does not cool the gas after it enters the swirl chamber through the nozzles. The gas in the preferred embodiment is cooled as it passes through the adiabatic nozzles, but it will be understood, of course, that advantage from the invention can be obtained by cooling the gas in any other suitable means, for example by one of the turbo-expanders known in the art, and then passing the gas at velocity in excess of that of sound directly to the swirl chamber for separation into hot and cold components. The important feature to consider, however, is that the function of the swirl chamber is to sort out gas or vapor molecules into two separate fractions, one with high and the other with low kinetic energies. The colder molecules with low kinetic energies remain colder while those with high kinetic energies become heated through degradation of kinetic energy into thermal energy and are exhausted as the hot fraction of the gas. In the embodiments illustrated the inlet channels are so constructed that together with the entrance opening they function as adiabatic nozzles.

Figure 1 is a flow sheet of a throttling unit which may be of either of the types shown, applied to cooling a refrigeration load, in this case a water bath. The water bath **54** is contained within a suitably closed tank **55** which includes the water bath and air or other gas or vapor. Inlet for air and water vapor to the system is provided by a pipe **56′** from an inlet chamber **56** to a compressor **57** driven in any suitable manner by a power unit **58** at a suitable speed to run the compressor efficiently. In the wall of the inlet chamber is provided a series of ceramic plugs **59** of any well known character (as for example the Selas type), porous to water but to a limited extent impermeable to air. The porous plugs **59** are immersed beneath the water of the water bath and permit flow of water therethrough back and forth between the water bath and the inlet, while retaining the bulk of the air in the inlet. Each plug is connected on the inside to the inlet.

From the compressor the air and accompanying water are conducted through a pipe **60** to a water eliminator **63** of any well known type which may if desired be provided with a cooling coil **64** so that it can function as a cooler or preliminary intercooler if necessary. Water taken off by the water eliminator from the compressed air is disposed of in any suitable way as by manual removal, but preferably by drainage through a return pipe **65** to the water bath so that it can return and replenish the water in the water bath. It will be understood that reasonably good efficiency in water elimination is desirable to obtain high efficiency in the throttling device. From the water eliminator the compressed air is carried through a cooler **66** whose operation will be described, and from the cooler the cooled compressed air passes through a throttling device **67**, in accordance with the invention, and which may be of any of the types shown. From the throttling device **67** the colder fraction is carried through an exit **68** while the hotter fraction, after passing through the constriction, leaves by the exit **69**. In order to permit advantage from the fact that the hotter fraction has been deprived of part of the water vapor content, it is desirable to cool the hotter fraction in an after cooler **70** having a cooling coil **73** to approximate the ambient temperature. The hotter and colder fractions are then combined at **74** and the combined fractions are first carried through the jacket portion **75** of the cooler **66**, using part of the refrigeration to cool the inlet compressed air to the throttling device. From the cooler **66** the remainder of the combined hotter and colder fractions are carried through pipe **76** back to the inlet, where the air is again available to go through the next cycle.

Excess air may be drawn into the system by leaks at low pressure points which would result eventually in building up an excessive pressure. A suitable blow-off device **77** is provided on the high pressure side to eliminate air which might increase the inlet pressure above several inches of mercury. It is found to be cheaper and simpler, rather than using a pressure blow-off valve of any of the well known types, to employ a well-known plug at **77** which is porous to air and impermeable to water (for example of the Selas type) causing a constant slight leakage outwardly which will make up for inlet leakage at other points.

In operation it will be understood that air accompanied by water vapor passes through the compressor and is compressed, and the heat of condensation of the water is absorbed by the cooling medium in the water eliminator **63** or the cooler **66** or both, returning the water to the water bath as liquid water. It is not necessary to go to the expense of making an air tight closed cycle, and the air leaking into the system is utilized as effectively as possible. The air pressure in the water bath is preferably held at a level less than two inches of mercury, the excess air being ejected from the system, as for example through the blow-off means **77**.

In order to evaporate water in the water bath heat of evaporation must be supplied. In Figure 1 this heat is supplied from the water of bath **54**. In the water eliminator **63** part of this water is condensed and returned to the water bath by the pipe **65**, which returns only water, and not any appreciable amount of compressed air. The heat given off on condensing water vapor is taken away by cooling coil **64** so that the net result is that the water bath has to supply heat and thus cool itself in the evaporation process. The water bath is also cooled by the cooled air returning by pipe **76**. The return of condensed water back to the water bath prevents depletion of the water in the water bath.

In view of my invention and disclosure variations and modifications to meet individual whim or particular need will doubtless become evident to others skilled in the art, to obtain all or part of the benefits of my invention without copying the process and structure shown, and I, therefore, claim all such insofar as they fall within the reasonable spirit and scope of my claims.

Having thus described my invention what I claim as new and desire to secure by Letters Patent is:

1. In a refrigerating device, a compressor, a heat exchanger connected to the high pressure side of the compressor, walls forming an open interior generally circular swirl chamber which is at least in part stationary, inlet means including a nozzle having an abruptly converging portion, a throat and a diverging portion having a length about five times the difference between the throat diameter and the mouth diameter for introducing gas or vapor from the compressor into the swirl chamber in a swirl whose axis conforms to the axis of the chamber, whose peripheral velocity exceeds the velocity of sound for the particular gas or vapor and whose pressure at the outside exceeds the pressure at the axis by at least one-half atmosphere, the mouth being at least as close as three times the mouth diameter to one end of the chamber, walls forming exits extending endwise of the chamber for hotter and colder fractions, the exit for the colder fraction being through said end wall and relatively smaller and closer to the axis and the exit for the hotter fraction being relatively farther from the axis, a connection for carrying at least a part of the colder fraction into heat transfer relation with the heat exchanger and a constriction in the exit for the hotter fraction beyond the chamber.

2. A device according to claim 1, in which the walls forming the open interior swirl chamber are entirely stationary.

3. A device according to claim 1, in which one of the walls forming the open interior swirl chamber is stationary and another of the walls comprises a rotor rotated by the gas or vapor.

4. A device according to claim 1, in which the open interior swirl chamber is of conical form, with outlet for the cold fraction at the small end and for the hot fraction at the large end.

5. A device according to claim 1, in which the open interior swirl chamber is conical, and the upper wall thereof comprises a rotor which is turned by the gas or vapor and floats on the gas or vapor swirl.

6. A device according to claim 1, in which the exit for the colder fraction and the exit for the hotter fraction are in opposite ends of the swivel chamber.

7. A device according to claim 1, in combination with means for recycling a portion of said hotter fraction back to the inlet of the compressor.

8. A device according to claim 1, in combination with a supplementary cooling unit and means for recycling said hotter fraction through the supplementary cooling unit and thence back to the inlet of the compressor.

9. A device according to claim 1, in combination with means for conveying the colder fraction back to the inlet of the compressor after it passes in heat transfer relation with the heat exchanger.

10. A device according to claim 1, in combination with means for removing condensed vapor from the fluid stream after it leaves the outlet of the compressor and before it enters the swirl chamber.

11. In a refrigerating device, a compressor, a heat exchanger connected to the high pressure side of the compressor, walls forming an open interior generally circular swirl chamber which is at least in part stationary, inlet means including a nozzle having an abruptly converging portion, a throat and a diverging portion having a

length about five times the difference between the throat diameter and the mouth diameter for introducing gas or vapor from the compressor into the swirl chamber in a swirl whose axis conforms to the axis of the chamber, the mouth being at least as close as three times the mouth diameter to one end of the chamber, walls forming exits extending endwise of the chamber for hotter and colder fractions, the exit for the colder fraction being through said end wall and relatively smaller and closer to the axis and the exit for the hotter fraction being relatively further from the axis, a connection for carrying at least a part of the colder fraction into heat transfer relation with the heat exchanger and a constriction in the exit for the hotter fraction beyond the chamber.

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