SUPERINTENDENCIA DE INDUSTRIA Y COMERCIO
DIRECCION DE NUEVAS CREACIONES

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Abogado
Sr. JORGE ARTURO ROMERO PRIETO.

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	Trámite	002
	Evento	001
	Actuación	430
	Oficio	2532
	Folios	11

Patente de Invención [ ] Patente de Modelo de Utilidad [X]

Vía Nacional [X]

Vía PCT (fase nacional) [ ] Cap.I [ ] Cap. II [ ]

No. Sol. Internacional ____________
Fecha de Presentación Internal. ____________

Como resultado del examen de fondo, realizado con fundamento en el artículo 85 y en concordancia con el artículo 45 de la Decisión 486 de la Comisión de la Comunidad Andina, se le comunica:

1. Documentos estudiados

Descripción:	Folios:	8	a	13	Como se radico inicialmente.
Reivindicaciones:	Folios:	14	a	15	Como se radico inicialmente.
Dibujos:	Folios:	17	a	19	Como se radico inicialmente.

2. Objeto de la invención

Titulo de la invención: **"COMPRESOR PARA GNV CON FILTRACION Y TEMPERATURA CONTROLADA". (Folio 1).**

Problema Técnico Planteado:

- Aumentar la cantidad de gas combustible al suministrar un vehículo para su respectivo funcionamiento, evitando que al comprimir el gas en una unidad de compresión, su temperatura se incremente, aumentando la dificultad en su manejo y seguridad debido a su estabilidad molecular por las condiciones físicas a la que esta expuesto, la cual repercuten en la cantidad del gas combustible al suministrarlo.

- Minimizar el suministro de gas combustible que contengan material particulado, impurezas, contaminación o retención de humedad, antes de que el gas combustible llegue al motor del vehículo, evitando daños posteriores a los componentes del motor y su respectivo funcionamiento.

Solución:

- La solución al problema técnico planteado es proporcionar una nueva instalación de unidad compresora para gas natural vehicular (GNV) la cual incluye una fase de medición y filtración controlada con base en un filtro de doble efecto es decir un filtro de partículas y coalescencia o retenedor de humedad durante la compresión del gas. Este compresor consta de 3 a 4 etapas según la presión de succión y toma el gas directamente del gasoducto a presión de entre 863 y 2700 Kpa, y durante la compresión en sus diferentes etapas, somete al gas a cambios de temperatura por efecto de la fricción molecular, que pueden estimarse entre los 90 y los 130 grados centígrados.

IPC: **F04D27/02, F04D27/00, F04B49/00.**

3. Análisis de la solicitud de Patente

Reivindicaciones *(artículo 30 D 486, Art 22 PCT).*

Claridad:

- En la reivindicación independiente No.1 presentan las siguientes falencias:

  - En esta reivindicación se expone *"Un motor eléctrico acoplado al compresor"*, aunque es muy elemental esta disposición entre estos elementos, esta disposición no se encuentra, ni se evidencia, ni se enuncia, por lo tanto esta disposición y característica técnica no se encuentra soportada en el capitulo descriptivo de la solicitud en estudio.

  - En esta reivindicación se ilustra la característica técnica "*Un sistema para contrarrestar pulsaciones emanadas por el compresor durante su funcionamiento"*, se evidencia que esta característica no es clara, debido a que no se sabe si se esta refiriendo a la misma característica expuesta en la descripción que dice *"Sistema de amortiguador de pulsaciones (11)"* originando confusión y mal entendimiento al momento de comprender esta reivindicación.

  - Al igual que el punto anterior también se presenta el mismo caso con la característica técnica que dice *"Intercambiadores de calor o temperatura"* se evidencia que esta característica no es clara, ni concreta, debido a que no se sabe si se esta refiriendo a la misma característica expuesta en la descripción que dice *"intercambiador de temperatura (9)"*, además esta característica técnica esta mal redactada, debido a estas falencias, igualmente se origina confusión y mal entendimiento al momento de comprender esta reivindicación.

  - No es clara la característica técnica *"sistema o estación de regulación y medición"* que se expone en esta reivindicación, debido a que no se sabe si se esta refiriendo a la misma característica expuesta en la descripción que dice *"una etapa (8) de medición y regulación controlada"*.

Se le sugiere al solicitante presentar una nueva reivindicación independiente No.1, que contenga en el preámbulo los elementos conocidos en el estado de la técnica y en la parte caracterizadora presentar los elementos y/o características técnicas esenciales novedosas que lo hacen diferente del estado de la técnica.

Se le recuerda al solicitante que en la reivindicación independiente, la parte caracterizadora, que suele seguir al preámbulo precedida por la frase ***"caracterizado por"***, recoge las características técnicas que van hacer protegidas y que constituyen la novedad frente al objeto conocido del estado de la técnica.

Se le sugiere al solicitante que al presentar una nueva reivindicación independiente No.1, colocando los nombres de los elementos y/o componentes correctos guardando correspondencia con los elementos y/o componentes que se describen en el capitulo descriptivo.

- En los siguientes grupos de reivindicaciones independientes y dependientes, se evidencia que ilustran elementos, características, componentes o términos, que no son claros, no se definen de manera especifica y concreta a que elementos se están refiriendo y/o cuales elementos pertenecen, así:

    - No.1. "Batería de cilindros", "Tropicalizado".
    - No.2. "Filtro de doble efecto".
    - No.3. "Medios resortados repartidos".
    - No.3. "Plataforma al piso".
    - No.3. "Sistema antivibratorio".
    - No.4. "Sistema para contrarrestar pulsaciones".
    - No.4. "Recipiente o contenedor".
    - No.4. "Bafles o desviadores de las pulsaciones".
    - No.6. "Controlador lógico programable o PLC".
    - No.7. "Juego de sensores", "Instalación".
    - No.8. "Dispositivo variador de velocidad".

    Se le sugiere al solicitante que coloque de manera especifica y concreta a que elementos, características, componentes o términos, se esta refiriendo, colocando al frente de la característica o del elemento, el numero de referencia guardando correspondencia con el capitulo descriptivo, las graficas o dibujos y dicho capitulo reivindicatorio, con el objetivo de comprender mejor la invención al momento de hacer el análisis y distinguir los elementos de dicha invención.

- En la reivindicaciones dependientes No.5 y No.6, se evidencia que no están ilustrando de manera clara, las características técnicas particulares que conforman el *"compresor para GNV con filtración y temperatura"* de la solicitud en estudio, debido a que en estas reivindicaciones se esta ilustrando es el funcionamiento y la utilización del *"dispositivo controlador programable o PLC"*.

    Se le sugiere al solicitante redactar nuevamente las reivindicaciones dependientes No.5 y No.6, colocando de manera clara, las características técnicas particulares del *"compresor para GNV con filtración y temperatura"*.

- En la reivindicación dependiente No.8, se ilustra el elemento *"dispositivo variador de velocidad programable"*, se evidencia que esta característica o elemento no es claro, debido a que no se encuentra, ni se evidencia, ni se enuncia, en el capitulo descriptivo de la solicitud en estudio.

  Se le sugiere al solicitante redactar nuevamente la reivindicación dependiente No.8, colocando de manera clara, las características técnicas particulares del *"compresor para GNV con filtración y temperatura"*.

  Se le sugiere al solicitante que al presentar el nuevo capitulo reivindicatorio coloque los nombres de los elementos y/o componentes correctos guardando correspondencia con los elementos y/o componentes que se describen en el capitulo descriptivo.

  Se le recuerda al solicitante que los funcionamientos, usos y/o ventajas no son características patentables.

  Se le sugiere al solicitante, que en el capitulo reivindicatorio coloque los números de referencia al frente de todos los componentes y/o elementos que hacen parte o conforman la invención, guardando una correspondencia entre las graficas y/o dibujos, el capitulo descriptivo y dicho capitulo reivindicatorio, con el objetivo de comprender mejor la invención al momento de hacer el análisis, y a su vez distinguir los elementos de dicha invención.

Descripción *(artículo 28 D 486)*.

Claridad:

- En la capitulo descriptivo de la solicitud se presentan las siguientes falencias:

  - Los siguientes elementos llamados *"Bafles desviadores"*, *"Compresor" "intercambiador de temperatura (9)"*, y *"Ventilador"*, no se ilustran de manera clara en las graficas o dibujos.

  - Los elementos ilustrados con los números No.2, 3 y 5 en los dibujos o graficas, no están soportados en el capitulo descriptivo.

  - No se ilustra de manera clara el elemento con el número de referencia (8) que dice "Etapa (8) medición o regulación", tal como esta redactado no se sabe si es un elemento o una etapa.

  - En el capítulo descriptivo no se muestra claramente un orden lógico de la secuencia de funcionamiento y como interactúan entre si todos los componentes que conforman el *"compresor para GNV con filtración y temperatura"*.

  Se le sugiere presentar un nuevo capitulo descriptivo corrigiendo todas las falencias enunciadas anteriormente, y se le recuerda que no puede ampliar la materia de la invención cuando presente el capítulo descriptivo.

4. Determinación del Estado de la Técnica

Teniendo en cuenta la fecha de presentación nacional, con fecha del 03 de Febrero del 2005, se encontraron los siguientes documentos relevantes:

Nº	Documento	Título	Fecha de Publicación
D1	US5290142	METHOD OF MONITORING A PUMPING LIMIT A MULTISTAGE TURBOCOMPRESSOR WITH INTERMEDIATE COOLING	01/03/1994
D2	US3441200	GAS COMPRESION SYSTEM HAVING INLET GAS CONTROL	29/04/1969

Los documentos más cercanos de acuerdo al estado de la técnica que se encontró es **D1 US5290142**, **D2 US3441200,** con respecto a la solicitud en estudio, la cual se relaciona y se anexa dentro del concepto técnico, debido a que según como esta reivindicada la solicitud en estudio no se puede elaborar el examen de patentabilidad debido a que, no presenta de manera clara las características técnicas esenciales, igualmente algunas de estas características no esta soportadas en su capitulo descriptivo, asimismo no presenta en forma clara y concisa lo nuevo que esta aportando a la tecnología, además presentan muchas falencias en su capitulo reivindicatorio.

En cumplimiento del artículo 85 en concordancia con el artículo 45 de la Decisión 486 de la Comisión de la Comunidad Andina, se le indica que debe dar respuesta dentro del término de treinta días contados a partir de la fecha de notificación de la presente comunicación. En caso de no dar respuesta al presente requerimiento, o si a pesar de la respuesta subsistieran los impedimentos para la concesión, se denegará la patente.

Notifíquese el presente oficio conforme a lo dispuesto en el numeral 5.2 literal e), del capitulo quinto del titulo primero de la circular única, No. 10 del 2001.

ALIX CESPEDES DE VERGEL
Jefe de División de Nuevas Creaciones

El anterior requerimiento fue notificado por fijación en lista, de fecha 3 FEB. 2010

CONCEPTO TECNICO Numero: 349	**EXAMINADOR TECNICO** Código No. 202072

22

US005290142A

United States Patent [19]

Ispas et al.

[11] **Patent Number: 5,290,142**

[45] **Date of Patent: Mar. 1, 1994**

[54] **METHOD OF MONITORING A PUMPING LIMIT OF A MULTISTAGE TURBOCOMPRESSOR WITH INTERMEDIATE COOLING**

[75] Inventors: **Ioan Ispas**, Kön; **Ulrich Grundmann**, Voerde; **Yvan Van Hoof**, Kön, all of Fed. Rep. of Germany

[73] Assignee: **Atlas Copco Energas GmbH**, Cologne, Fed. Rep. of Germany

[21] Appl. No.: **952,964**

[22] Filed: **Sep. 29, 1992**

[30] **Foreign Application Priority Data**

Oct. 1, 1991 [DE] Fed. Rep. of Germany 4132735
Jan. 28, 1992 [DE] Fed. Rep. of Germany 4202226

[51] **Int. Cl.**[5] **F04D 27/02**

[52] **U.S. Cl.** **415/1**; 415/11; 415/15; 415/17; 415/27; 415/47

[58] **Field of Search** 415/1, 11, 15, 17, 26, 415/27, 47; 417/253

[56] **References Cited**

U.S. PATENT DOCUMENTS

3,424,370	1/1969	Law	415/1
3,441,200	4/1969	Huesgen	415/47
4,288,198	9/1981	Hibino et al.	415/1
4,362,462	12/1982	Blotenberg	415/1
4,810,163	3/1989	Blotenberg	415/17
4,831,534	5/1989	Blotenberg	415/15
4,831,535	5/1989	Blotenberg	415/1
4,944,652	7/1990	Blotenberg	415/1

FOREIGN PATENT DOCUMENTS

235697	9/1988	Japan	415/17
964251	10/1982	U.S.S.R.	415/17
987193	1/1983	U.S.S.R.	415/17
1366713	1/1988	U.S.S.R.	415/1

Primary Examiner—Edward K. Look
Assistant Examiner—James A. Larson
Attorney, Agent, or Firm—Herbert Dubno

[57] **ABSTRACT**

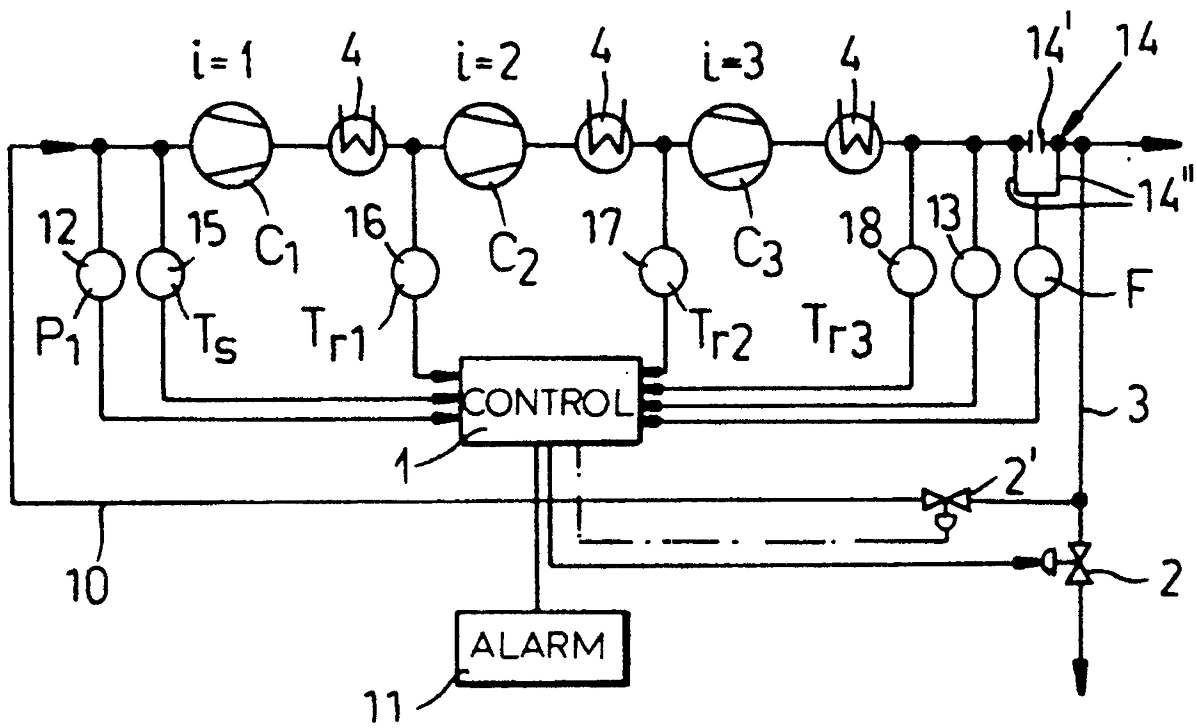
A method of operating a multistage turbocompressor or of monitoring the pump limit thereof, whereby the pump limit function is stored in the control unit in the form $Y=m.F+b$ and the coefficients m and b are determined by the linear relationships

$$m=m_0+m_1.dT_s+\Sigma m_{2i}.dT_{ri}$$

$$b=b_0+b_1.dT_s+\Sigma b_{2i}.dT_{ri}$$

where T_s and T_{ri} represent the intake temperature and the backcooling temperatures following coolers after each turbocompressor and dT_s and T_{ri} and reference temperatures T_{ref}. The system automatically compensates for the shift of the pump limit with temperature and allows operation closer to the pump limit or boundary for stable aerodynamic operation.

5 Claims, 4 Drawing Sheets



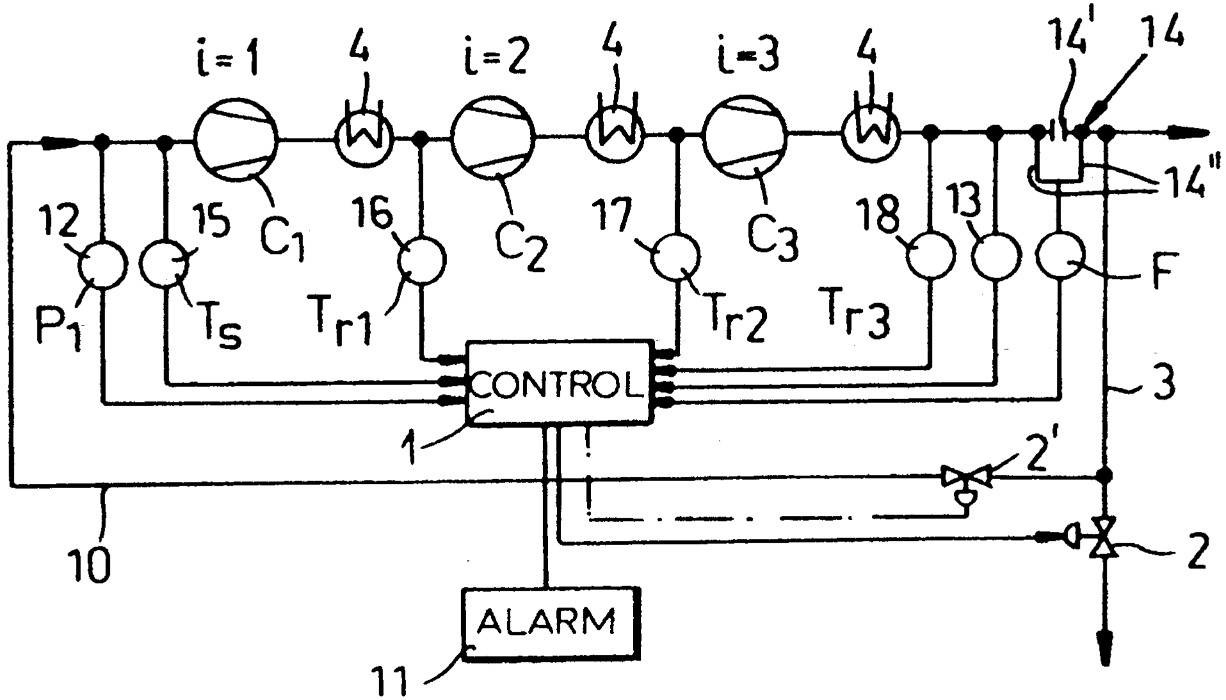

i=1
4
i=2
4
i=3
4
14'
14
14"
12
15
C1
16
C2
17
C3
18
13
P1
Ts
Tr1
Tr2
Tr3
F
CONTROL
3
1
2'
10
2
ALARM
11

FIG.1

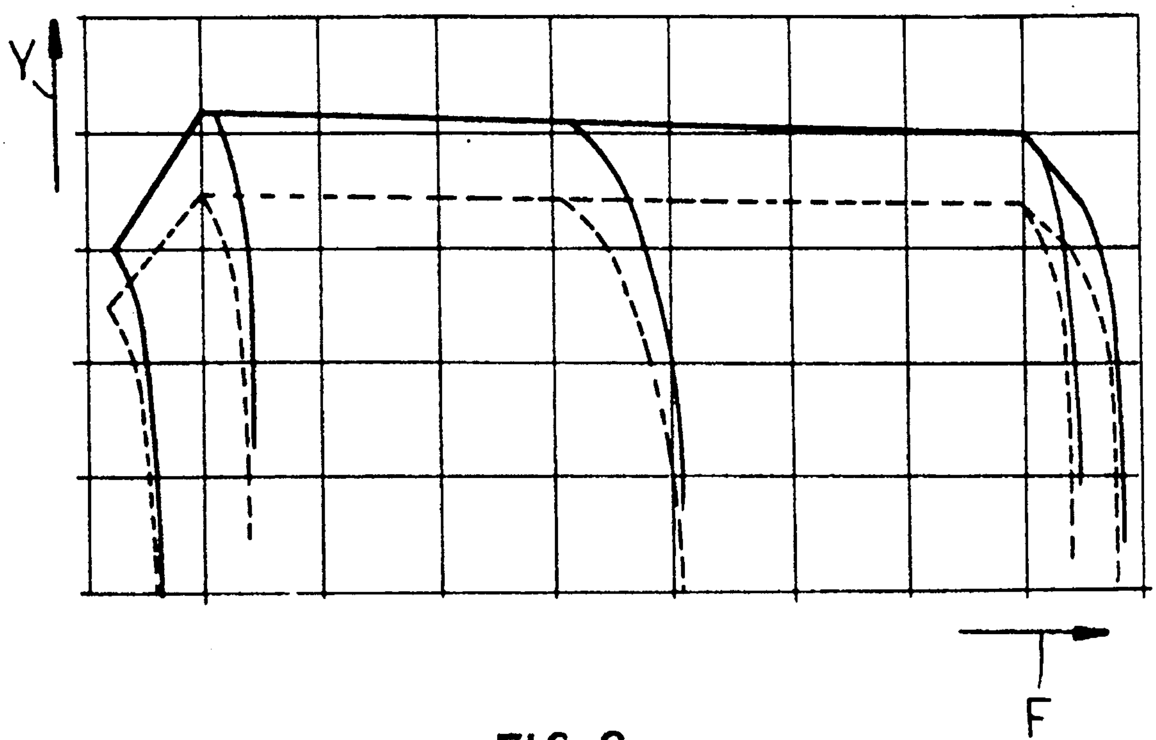


FIG. 2

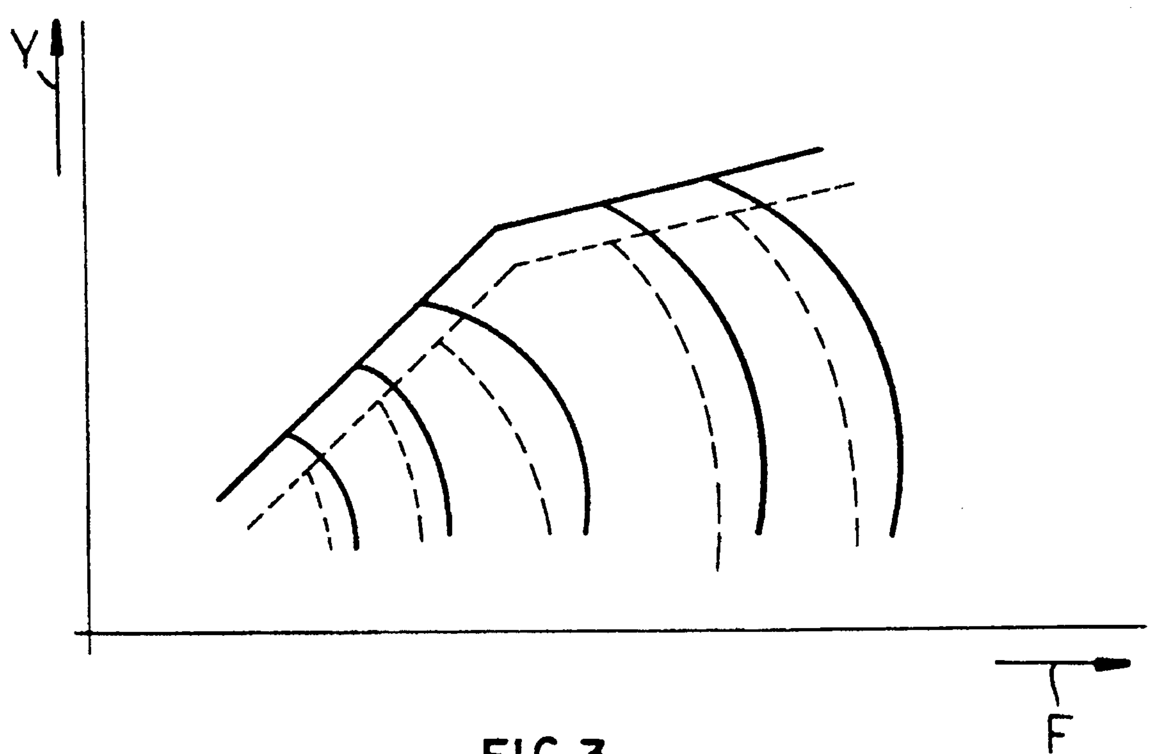
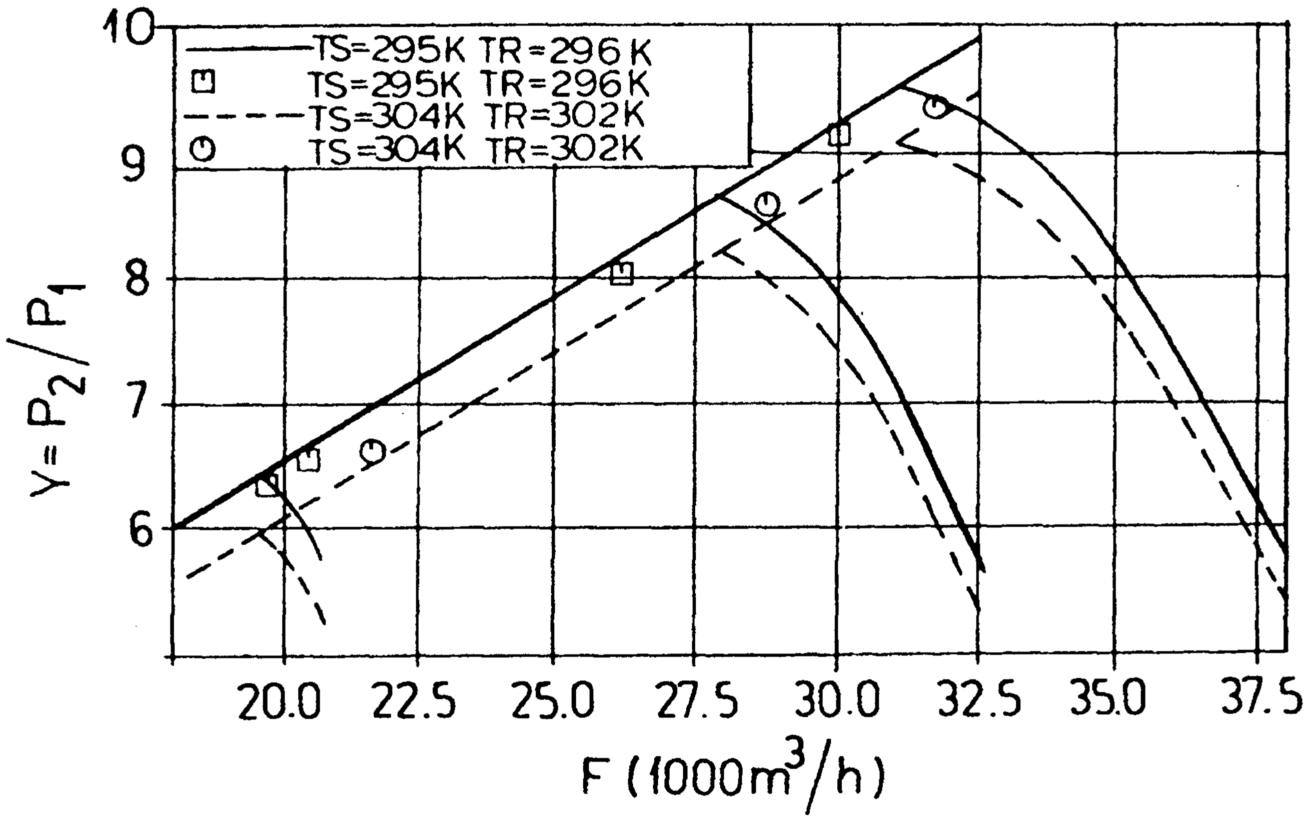


FIG. 3



TS=295K TR=296K
TS=295K TR=296K
TS=304K TR=302K
TS=304K TR=302K
10
9
8
7
6
Y = P2/P1
20.0
22.5
25.0
27.5
30.0
32.5
35.0
37.5
F (1000m3/h)

FIG.4

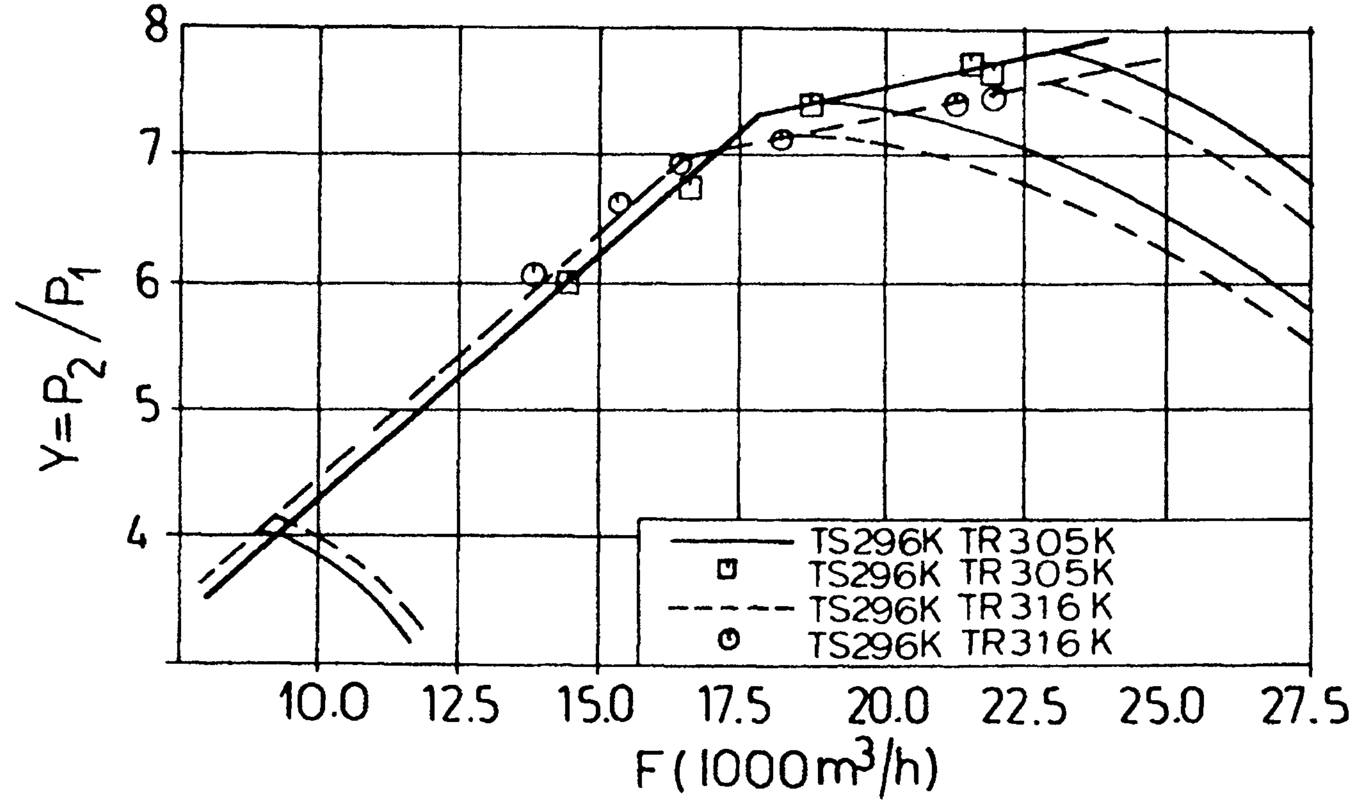

8
7
6
5
4
$Y=P_2/P_1$
10.0
12.5
15.0
17.5
20.0
22.5
25.0
27.5
F(1000m³/h)
TS296K TR305K
TS296K TR305K
TS296K TR316K
TS296K TR316K

FIG.5

24

METHOD OF MONITORING A PUMPING LIMIT OF A MULTISTAGE TURBOCOMPRESSOR WITH INTERMEDIATE COOLING

FIELD OF THE INVENTION

Our present invention relates to a method of monitoring the pumping limit of a multistage turbocompressor, i.e. a turbocompressor cascade, with intermediate or intervening cooling, i.e. cooling between the stages or following each turbocompressor stage. More particularly, the method of the invention relates to the operation of a multistage turbocompressor with postcompression cooling and deals specifically with the problem of potential operation at a pumping limit of the operating graph for the cascade.

BACKGROUND OF THE INVENTION

A turbocompressor cascade comprises a plurality of turbocompressors, i.e. compressor turbines, between which a heat exchanger is provided for the intermediate cooling of the compressed medium, generally a gas, so that the compressed gas passing to the next compressor stage is after cooled. Generally a corresponding heat exchanger is provided following the final stage as well.

The parameters of such a turbocompressor cascade can include the intake side pressure (p_1) the discharge side pressure (p_2), the intake side temperature or suction temperature T_s, the intermediate temperatures (or after-cooling temperatures) following cooling T_{ri}, where i represents the stage, i.e. is 1, 2, 3 . . . , n, also referred to as backcooling temperatures, and the volume rate of flow (F) through the system.

The measured values (p_l, p_2, F, T_s, T_{ri}) are fed to a control device with a data storage capability, can be compared with values of a compressor characteristic operating graph (i.e. the empirically derived optimum operating characteristics in a form enabling such storage and comparison) so that when the measured values of p_1, p_2 and F, for example, approach limits of the compressor operating graph, i.e. the so-called pumping limits, a warning signal or controlled signal is provided to prevent continued operation at or beyond these limits or, at least, to alert operating personnel that the limits have been approached.

The reference to a compressor operating graph is intended to include a collection of stored data in any form enabling the comparison of the measured values with corresponding empirically determined conditions of intended operation capable of indicating the approach to the pumping limit. For example, the "graph" can simply be tabulated data or operating tables derived from such data or other parameters automatically calculated by computer from stored data. All of these techniques are known in the art.

The pumping limit is defined as an aerodynamic stability boundary for the operating graph which limits the utility of the turbocompressor. When operation is effected below the pumping limit, refluxing or backflow can occur in the turbocompressor which results in pressure fluctuations, temperature increases and significant increase in the pumping noise.

Operating below these limits or at the pumping limit can lead in very short order to bearing failure and damage to the turbine blades.

To avoid operation of a turbocompressor in an unstable range, i.e. at or below the pumping limit, monitoring and controlled devices are provided which, upon critical approach to or passage of the pumping limit, open blow-off valves to the atmosphere or open a bypass valve in a circulating line which connects the pressure and suction sides of the turbocompressor. In this manner, a minimum flow through the turbocompressor is maintained and operation below the pumping limit is precluded.

Intermediate-cooled turbocompressors should allow control over a wide volume range. Control elements for this purpose can include inlet or outlet guide devices or drives with variable speeds for the turbocompressors. Combinations of these controlled elements can also be provided. However, the full utilization of the entire operating graph of the turbocompressor cascade requires that the pumping limit be determinable with precision.

Conventional processes for monitoring the pump limit of multistage intermediate cooled turbocompressors ignore the influence of changes in the cooling temperature on the pumping limit. The intermediate cooling temperature, constituting the temperature of the gas at the input to each successive stage has been found to play a major role upon the location of the pumping limit in the operating field of the turbocompressor.

In operation of the turbocompressor, variations of the stage input temperatures cannot be avoided and these variations can be a function of soiling of or poorly operating coolers, variations in the flow of the cooling agents through the heat exchangers or the like.

There are also seasonal changes in the intake temperatures of the gases to be compressed.

Ignoring the temperature effect on the location of the pumping limit requires that the turbocompressor be operated within much narrower operating limits with respect to volume variations, since the approach to the actual pumping limit can never be guaranteed entirely. Thus conventional systems of this type operate with limited versatility and with loss of range.

OBJECTS OF THE INVENTION

It is, therefore, the principal object of the present invention to provide a method of monitoring the operation of a turbocompressor with intermediate cooling in which the aforementioned problems are avoided.

A more specific object of the invention is to provide a method of operating a multistage turbocompressor with interstage cooling whereby the effect of the gas temperature at the intake of each stage is considered with respect to the position of the pump limit, so that the apparatus can be operated closer to that limit and thus with a wider range of variability with respect to control of the volume, etc.

Still another object of the invention is to provide an improved turbocompressor system of the aforedescribed type whereby the versatility thereof is enhanced.

SUMMARY OF THE INVENTION

These objects are attained, in accordance with the present invention, by measuring the intake temperature T_s at the intake of the first compressor stage of the turbocompressor as well as each of the cooldown temperatures following the respective heat exchanger following each turbocompressor stage and represented at T_{ri} (where i=1 to the number of cooling stages following respective turbocompressor), feeding the temperature measurement values to the control device and generating temperature differences dT_s, dT_{ri} from a predetermined reference temperature T_{ref}, determining an operating point of the turbocompressor with respect to a pumping limiting value based upon a pumping limit function $Y=m.F+b$ stored in the controlled unit and which represents a (linear) characteristic of the pumping limit in the compressor operating graph at least in part.

Y represents either the pressure ratio (P_2/P_1) between the discharge side pressure p_2 and the intake side pressure p_1 of the cascade or a pressure difference (P_2—P_1) between the pressures p_1 and p_2. The coefficients m and b are linear functions of the intake temperature T_s and the backcooling temperatures T_{ri} and are determined by use of the aforementioned temperature differences dT_s and dT_{ri} based upon the relationships

$$m=m_0+m_1.dT_s+\Sigma m_{2i}.dT_{ri}$$

$$b=b_0+b_1.dT_s+\Sigma b_{2i}.dT_{ri}$$

The volume flow rate and measured pressure values (F_1, P_1, P_2) are compared with the pumping limit value and upon passing below a predetermined minimum distance therefrom, trigger a warning or control signal.

For measuring the volume rate of flow in a pipe at a location along the cascade, we may make use of the generally accepted method of determining a pressure differential across a standard aperture, nozzle, diaphragm or venturi port. It will be understood that the volume flow rate can be obtained in terms of the measured pressure and temperatures across the aperture, orifice, diaphragm and the venturi nozzle and that the value of the pressure difference can be used directly as the input to the controlled unit. It should be understood further that different reference temperatures for the intake temperatures and backcooling temperatures can be used with the reference temperatures being selected to be appropriate to the thermodynamic design of the turbocompressor cascade. These may be determined empirically without difficulty as well.

The operating graph or range of parameters for the compressor for different intake and backcooling temperatures including the pump limit can be determined by superimposition of the characteristic curves for the individual stages. The calculation of the turbocompressor graphs is conventional in the art and the pumping limits are approximated by straight lines for all of the calculated cases The parameters m_o, m_1, m_2, b_o, b_l, b_2 are generally calculated by solving systems of linear equations. Changes of the stage intake temperatures (dT_s, dT_{ri}) primarily effect the coefficient b. That results in a shifting of the pump boundary or limit in the operating field. The temperature effect on the slope of the pump limit function is substantially smaller so that, in many cases,

$$m_1=m_2=0$$

and the temperature effect on the slope of the pump limit function is negligible.

If the backcooling temperatures differ only slightly from one another, it is sufficient to provide as the measured value of the backcooling temperature (T_{ri}) an arithmetic mean over the compressor stages (i=1, 2 . . . n), the mean temperature being represented at T_r and being utilized to calculate the coefficients m and b. In this case, the functions for calculating the coefficients m and b can be simplified to the following:

$$m=m_0+m_1.\ dT_s+.dT_r$$

$$b=b_0+b_1.\ dT_s+.dT_r$$

In a preferred embodiment of the invention the temperature coefficients m_1, b_1, m_{2i}, b_{2i} or m_2, b_2, calculated by variation of the intake and backcooling temperatures by the thermodynamic operating field calculations for the compressor cascade, are stored as constants in the control device, while the coefficients m_0 and b_0 are determined empirically by field tests on the installed turbocompressor cascade and are then inputted to the control computer. The pump limit function can then be determined with great precision and simultaneously calibrated. In order to calibrate the pump limit function, the turbocompressor can be driven briefly at its pump limit and the respective pressure, temperature and volume flow values measured and evaluated by the aforementioned equations the coefficients m_0, b_0.

More particularly, a method of operating a multistage turbocompressor system in which at an intake temperature T_s and pressure p_1 a gas is drawn into a first turbocompressor, gas compressed in the first turbocompressor is subject to intermediate cooling, cooled gas is compressed in subsequent turbocompressors in cascade with subsequent cooling to temperatures T_{ri} where i is a number which represents the number of the cooling stage in succession at which the cooling occurs, and the gas after a last turbocompressor and cooling stage of the cascade has a discharge pressure p_2 for a volume rate of flow F through the cascade, the method comprising the steps of:

(a) driving the cascade and measuring the intake temperature T_s, each temperature T_{ri}, temperature differences dT_s and dT_{ri} of the temperatures T_s and T_{ri} and a predetermined reference temperatures T_{ref}, the pressures p_1 and p_2 and the volume rate of flow F;

(b) calculating a pump operating unit function $Y=m.F+b$ relating the pressures p_1 and p_2 and the rate of flow F and describing a pump limit in a stored compressor performance graph, where Y is a pressure difference or ratio of the discharge pressure p_2 and the intake pressure p_1, and m and b are coefficients which are linear functions of the intake temperature T_s and temperatures T_{ri} and the temperature differences dT_s, dT_{ri} where;

$$m=m_0+m_1.dT_s+\Sigma m_{2i}.dT_{ri} \text{ and}$$

$$b=b_0+b_1.dT_s+\Sigma b_{2i}.dT_{ri},$$

where m_0, b_0, m_1, $b_1$and m_{2i}, b_{2i} are empirically determined constants;

(c) storing the pump operating limit function;

(d) comparing measured values of the volume rate of flow F and the pressures p_1 and p_2 of step (a) during driving of said cascade with corresponding values in the compressor storage graph; and

(e) generating a warning or control signal upon operation of the cascade with said measured values falling to a predetermined degree below a predetermined distance from the stored pump operating limit function.

The advantage of the invention is that the shifts in the pump limit as a consequence of changes in the intake and backcooling temperatures are automatically taken into consideration and the position of the pump limit at variations of the operating point of the turbocompressor can be determined with precision. The safety factor with respect to the pumping limit at which one operates can be greatly narrowed and the volume control range of multistage intermediate cooled turbocompressor can be more fully utilized. The equipment and systems required for carrying out the invention are not at all costly.

BRIEF DESCRIPTION OF THE DRAWING

The above and other objects, features and advantages of my invention will become more readily apparent from the following description, reference being made to the accompanying highly diagrammatic drawing in which:

FIG. 1 is a diagrammatic illustration of a turbocompressor cascade embodying the invention;

FIGS. 2 and 3 are graphs in which Y, the pressure difference or ratio as described above, and F being plotted respectively along the ordinate and the abscissa; and

FIGS. 4 and 5 are turbocompressor operating graphs showing appropriate values for a mean backcooling temperature T_r and intake temperature T_s for various values of Y, plotted along the ordinate, and F plotted along the abscissa and representing assemblies of how the invention can be carried out in practice, illustrating a calculated pumping limit on the one hand and a pumping limit determined by field tests in the manner described on the other hand.

SPECIFIC DESCRIPTION

The turbocompressor shown in FIG. 1 has three turbocompressors C_1, C_2 and C_3 for the three stages i=1, i=2, i=3 of the turbocompressor cascade and is used for the compression, for example, of air or nitrogen although this utility is not exclusive. The turbocompressor may be used ahead of or in conjunction with an air rectification unit for the separation of oxygen or nitrogen or both from air or, for that matter, whenever high degrees of compression may be required.

For each compression stage backcooling is effected, for example, by respective heat exchangers 4 for each stage, the heat exchangers being fed with cooling agents by means not shown and standard in the art.

The turbocompressor cascade has a control unit 1 with the capacity for data storage, e.g. a computer, which serves to monitor the pumping limit and can control a valve 2, which is normally closed and which, upon opening, can vent the system to the atmosphere. Alternatively, the control unit 1 can control a valve 2' connected in a bypass 10 between the pressure side of the compressor and the intake side thereof.

If the operating point of the compressor approaches the pump limit, the control 1 can open one or both of the valves 2, 2' and/or deliver a signal to an alarm 11 capable of warning the operator of the proximity to the pump limit and thus the advent of a potentially dangerous situation.

The turbocompressor cascade is provided with measuring devices 12 and 13 for measuring the intake pressure p_1 and the discharge pressure p_2 at the intake and discharge sides of the cascade. These measuring devices may be standard manometers or pressure gauges capable of outputting appropriate digital or analog signals representing the pressures to the control computer 1.

The system is also provided with a measuring device 14 for measuring the volume flow F. The latter device may be an orifice signified at 14' and means represented at 14" for determining the pressure differential across that orifice.

In addition, temperature measuring devices 15, 16, 17, 18 are provided for measuring the intake temperature T_s at the intake side of the first compressor stage m=1 and the backcooling temperatures T_{ri}, namely, T_{r1}, T_{r2} and T_{r3} following each compressor turbine. The backcooling temperature is defined as the temperature of the gas stream in the flow direction downstream of the heatexchanger for backcooling the gas after the heating thereof by compression in the respective turbocompressors. All of these values are fed to the control computer 1.

The control computer 1 stores a pump limit function Y=m.F+b which describes the substantially linear characteristic of the pump limit in the operating graph of the turbocompressor either entirely or in part.

The value Y corresponds to the pressure difference or the pressure ratio between the discharge side pressure p_2 of the turbo compressor cascade and the intake side pressure p_l thereof.

The coefficients m and b are linear functions of the intake temperatures T_s and the backcooling temperature T_{ri} and are determined by the relationships

$$m=m_0+m_1.(T_s-T_{ref})+\Sigma m_{2i}.(T_{ri}-T_{refi})$$

$$b=b_0+b_1.(T_s-T_{ref})+\Sigma b_{2i}.(T_{ri}-T_{refi})$$

In these relationships T_{ref} and T_{refi} are freely selectable reference temperatures. Most advantageously, however, the reference temperatures are predetermined design temperatures for the turbocompressor, established in the design of the turbocompressor from the expected thermodynamic characteristic thereof.

The operating graph of the compressor for different intake and backcooling temperatures, including the corresponding pumping limits, can be theoretically determined by superimposition of the individual stage characteristic or graphs. The pumping limits can be approximated by straight lines for all calculated cases and stored in the memory of the computer. The parameters m_0, m_1, m_{2i}, b_0, b_1, b_{2i} are determined by solving linear equation systems.

The graphs characterizing the turbocompressor operation and illustrated in FIGS. 2 and 3 show general cases in which the characteristic of the pump limit is at least in part a linear graph representing the pump limit function Y=m.F+b or which is approximated by the linear pump limit function Y=m.F+b. Changes in the intake and backcooling temperatures result in shifting of the pump limit as can be seen from a comparison of the broken and solid line positions in FIGS. 2 and 3.

FIGS. 4 and 5 illustrate the operating fields of two three-stage turbocompressors having at the intake side a control device regulating flow to the turbocompressor and serving as a volume control element for adjusting the operation of a range of volumes. The graph of the pump limit is completely approximated by the pump limit function (FIG. 4) or is approximated by the pump limit function over a portion of the graph (FIG. 5).

A comparison of the solid and broken line showings will demonstrate the temperature effect on the pump limit. In these graphs, moreover, values have been illustrated for the pump limit which were determined in field tests of the turbocompressors.

The comparison makes clear that the pump limit function stored in the control computer 1 defines the actual graph of the pump limit with great precision even with temperature variations of the gas stream and hence one can operate very close to the limit before the warning signal is triggered or the valves are opened. According to the pump limit function stored in the control device, an operating point of the turbo compressor can be determined by the measured pressure values p_1, p_2 or the measured volume value F as well as the measured temperature values T_s and T_{ri}. The measured values of the volume F and the pressures p_1 and p_2 are compared with this pumping limit and, upon falling below a predetermined minimum value of a distance from the pumping limit a warning signal or control signal for opening the valve 2 can be triggered. This minimum value can be a value representing the precision of the pumping limit as determined statistically.

We claim:

1. A method of operating a multistage turbocompressor system in which at an intake temperature T_s and pressure p_1 a gas is drawn into a first turbocompressor, gas compressed in the first turbocompressor is subject to intermediate cooling, cooled gas is compressed in subsequent turbocompressors in cascade with subsequent cooling to temperatures T_{ri} where i is a number which represents the number of the cooling stage in succession at which said cooling occurs, and the gas after a last turbocompressor and cooling stage of the cascade has a discharge pressure p_2 for a volume rate of flow F through the cascade, said method comprising the steps of:

(a) driving said cascade and measuring said intake temperature T_s, each temperature T_{ri}, temperature differences dTs and dT_{ri} of the temperatures T_s and T_{ri} and a predetermined reference temperatures Tref, the pressures p_1 and p_2 and the volume rate of flow F;

(b) describing a pump limit function $Y=m.F+b$ relating said pressures p_1 and p_2 and said rate of flow F in a stored compressor performance graph, where Y is selected from one of a pressure difference and a ratio of the discharge pressure p_2 and the intake pressure p_1, and m and b are coefficients which are linear functions of the intake temperature T_s and temperatures T_{ri} and the temperature differences dTs, dT_{ri} where:

$$m=m_0+m_1.dT_s+\Sigma m_{2i}.dT_{ri} \text{ and}$$

$$b=b_0+b_1.dT_s+\Sigma b_{2i}.dT_{ri}$$

where

m_0 and, b_0 are empirically determined constants and temperature coefficients m_1, b1, m2i, $b2_i$ are determined by thermodynamic calculations with variations of the intake and cooling temperatures and are stored in a control unit;

(c) operating the turbocompressor cascade at its pumping limit for the purpose of calibrating the pump limit function to yield pressure, temperature and volume values from which the coefficients m_0, b_0 for the pump limit function are determined and providing the coefficients m_0, b_0 as inputs to the control unit;

d storing said pump limit function;

e comparing measured values of the volume rate of flow F and the pressures p_1 and p_2 of step (a) during driving of said cascade with corresponding values in said compressor storage graph; and

f generating a control signal upon operation of said cascade with said measured values falling below a predetermined minimum value of a distance from said stored pump limit function.

2. The process defined in claim 1 wherein a warning signal is generated in step (f) to alert an operator to the passage of the measured values to a predetermined degree below said predetermined distance from said stored pump limit function.

3. The process defined in claim 1 wherein, in step (f), a vent valve at a discharge side of said cascade is opened.

4. The process defined in claim 1, further comprising opening a bypass valve connecting a discharge side of said cascade with an intake side thereof upon said measured values falling in step (f) to a predetermined degree below said pre determined distance from said stored pump operating limit value.

5. The process defined in claim 1 wherein an arithmetic mean is formed from said temperatures T_{ri} and the arithmetic mean is used to calculate the coefficients m and b.

* * * * *

27

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GAS COMPRESSION SYSTEM HAVING INLET GAS CONTROL

Eugene L. Huesgen, Jeannette, Pa., assignor to Carrier Corporation, Syracuse, N.Y., a corporation of Delaware

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U.S. Cl. 230—5 5 Claims

ABSTRACT OF THE DISCLOSURE

A gas compression system for supplying compressed air at a uniform pressure which employs a multistage compressor. A main control is provided to maintain the discharge pressure constant. The temperature and pressure of gas admitted to the first stage of the compressor and the temperature of gas admitted to the second stage of the compressor are sensed and functions thereof are summed to provide a control signal. An inlet gas throttling valve is modulated in response to the control signal to reduce overloading of the compressor and/or to keep the compressor from surging by varying the inlet air pressure.

BACKGROUND OF THE INVENTION

This invention relates to gas compression systems and more particularly to gas compression systems of the type employing a multistage axial or centrifugal compressor driven by a prime mover. A main control system may be provided which senses the discharge gas pressure and controls various system functions such as a discharge valve so as to maintain the discharge pressure of the system constant.

Two problems, however, are often experienced in attempting to maintain high operating efficiency with such systems, due to changing air temperatures and pressures. First, decreases in ambient air temperature or increases in ambient air pressure may increase the flow of air through the compressor, which tends to overload the prime mover employed to drive the gas compressor. The overloading of the prime mover may result in injury or destruction of it, and attempting to avoid an overload at all conditions at which may be encountered tends to establish an artificial limit on the range of conditions of operation of the gas compression system which results in reduced efficiency. Second, reductions in density of the ambient air admitted to the compressor, such as may occur with increasing ambient air temperature, may result in a reduction in mass flow through the compressor to the point that surge is encountered. Surge is a highly undesirable, unstable condition of the compressor which results in a loss of capacity and may cause serious damage to the compressor.

It has been proposed to employ a throttling valve in the line supplying inlet air to the compressor and to modulate the throttling valve in accordance with the density of the inlet air. This type of control provides greatly improved operation of the gas compression system over a range of inlet air conditions. However, such a control still does not provide optimum efficiency of operation because the operating characteristics of the system must be a compromise which is selected so that neither surge nor prime mover overload occurs at any anticipated condition. Furthermore, where an interstage gas cooler is employed in the operation of the system, this type of control must normally operate sufficiently away from the overload and surge regions so that changes in cooling fluid flow rate or temperature as well as variations in latent heat of the gas do not result in overloads or surging. Consequently, the optimum compressor operating efficiency and pressure lift cannot always be safely achieved due to the required compromise in operating characteristics.

SUMMARY OF THE INVENTION

In accordance with a preferred embodiment of this invention, a gas compression system is provided which employs a multistage air compressor having a suitable main control system to maintain a uniform discharge pressure. An inlet air throttling valve controls the pressure of air admitted to the first stage of the compressor. The temperature and pressure of inlet air as well as the interstage air temperature are sensed and compared against a reference signal to provide a control signal for varying the inlet air pressure by modulating the inlet throttling valve.

A control system in accordance with this invention provides improved operating efficiency and capacity, while at the same time it reduces the occurrences of overloads on the prime mover and surging of the compressor by taking into account the interstage gas temperature in controlling the operation of the system. Consequently, by employing the type of control system described herein, a gas compression system can be operated at more nearly optimum conditions over a wide range of ambient inlet air and interstage cooling conditions than previous systems.

BRIEF DESCRIPTION OF THE DRAWING

The figure schematically illustrates a gas compression system having a control system in accordance with this invention.

DESCRIPTION OF THE PREFERRED EMBODIMENT

This invention will be described with respect to a preferred embodiment thereof wherein the gas compression system is employed to supply compressed air to desired locations for operations of pneumatic equipment.

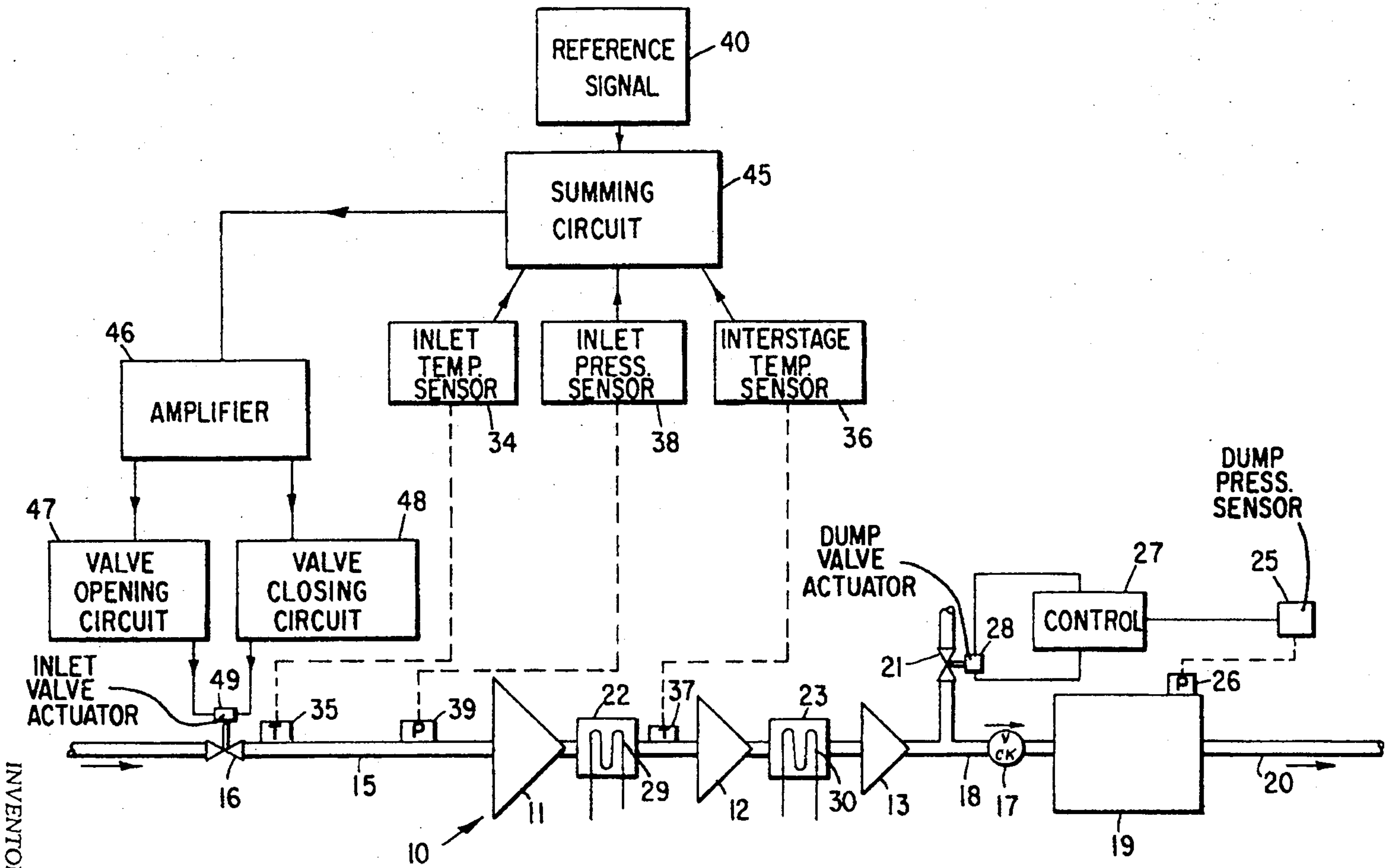
Referring particularly to FIGURE 1 there is shown a gas compression system embodying a multisage centrifugal or axial compressor **10** having a first stage **11** in series with a second stage **12** and a third stage **13**. Inlet gas line **15** is provided to admit ambient inlet air through a throttling valve **16** to first stage **11** of compressor **10**. Interstage gas coolers **22** and **23** having suitable cooling medium heat exchange passages **29**, **30** therein are provided to cool gas passing between the stages of compressor **10**.

Compressed air is discharged from compressor **10** through a check valve **17** in gas discharge gas line **18** into receiver **19**. A suitable compressed gas discharge line **20** distributes air from receiver **19** to the desired locations at which it is employed.

In order to maintain the pressure of discharge air in receiver **19** in line **20** relatively constant at a predetermined pressure, a dump valve **21** is employed to dis-

GAS COMPRESSION SYSTEM HAVING INLET GAS CONTROL

Filed March 13, 1967



INVENTOR.
EUGENE L. HUESGEN.
BY
Frank M. Becker Jr.
ATTORNEY.

20

charge compressed air from compressor **10** when the pressure demands of receiver **19** are satisfied. Dump valve **21** may be operated either as a fully opened or fully closed valve or it may be modulated according to the desired characteristics of the gas compression system.

A main control system is provided to control the operation of dump valve **21**. The main control system employs a discharge pressure sensor **25** at a suitable location **26** in line **18** or receiver **19**. A main control system **27** is actuated in response to a pressure signal generated by discharge pressure sensor **25** and sense a valve opening or closing signal to valve actuator **28**. Control system **27** maintains the discharge pressure at substantially predetermined value irrespective of the load imposed on the gas compression system.

In accordance with this invention, an additional control system is provided for controlling the inlet air pressure to first stage **11** of compressor **10** by modulating inlet gas valve **16** in accordance with the condition of operation of the system.

For this purpose, an inlet air temperature sensor **34** is provided at a suitable location **35** between inlet valve **16** and first stage **11** for sensing the temperature of air admitted to the first stage of the compressor.

An interstage temperature sensor **36** is also provided at a suitable location **37** between first stage **11** and second stage **12**. Interstage temperature sensor **36** is located to sense interstage gas temperature between the inlet of scond stage **12** and the outlet of interstage gas cooler **22**. Consequently, interstage temperature sensor **36** is able to compensate for variations in the flow rate and temperature of the cooling medium in the interstage cooler, as well as for unknown factors like scaling of the heat exchanger which may effect the actual interstage gas temperature.

An inlet air pressure sensor **38** is provided at a suitable location **39** between inlet gas valve **16** and first stage **11** for sensing the pressure of the air admitted to compressor **10**.

Inlet air temperature sensor **34** generates an inlet air temperature signal which is a known function of the inlet air temperature. For this purpose, sensor **34** may comprise a temperature responsive resistance element such as a thermistor. Because the actual characteristic of the desired inlet air temperature functions may not be linear or because the sensor may be nonlinear over the desired range of operating temperatures, a preferred embodiment of this invention employs a pair of thermistors in a suitable resistance network to provide a desired continuous, nonlinear temperature function signal. Interstage temperature sensor **36** may also comprise a thermistor in a resistance network which generates the desired interstage temperature function signal. Inlet pressure sensor **38** may comprise a pressure transducer in the form of a potentiometer having a wiping element movable in response to pressure variations and having the desired resistance characteristic to generate the required pressure function signal by dividing current flow in a differential amplifier.

A reference signal generator **40**, which may also comprise a potentiometer for dividing current flow in a differential amplifier, is employed to generate a reference signal corresponding to a predetermined desired condition of operation of the gas compression system, whereby the load on the compressor prime mover will not be excessive and the compressor will not be subject to surge.

The inlet temperature function signal, the interstage temperature function signal, the inlet pressure function signal and the reference signal are all summed in a summing circuit **45** to provide a control signal. The summing circuit serves to compare the sensed function signals with the reference signal to provide a difference or error control signal which indicates the need for adjustment of the inlet air pressure condition to maintain the desired condition of operation determined by the reference signal.

The control signal is passed to amplifier **46** which provides either an inlet valve opening signal or an inlet valve closing signal depending on the direction of error indicated by the control signal. The valve opening signal is passed, when present, to a valve opening circuit **47**, which in turn causes valve actuator **49** to open inlet gas valve **16**, thereby increasing the absolute pressure of air admitted to first stage **11** of compressor **10**. The valve closing signal, when present, is passed to valve closing circuit **48** which in turn causes valve actuator **49** to close inlet gas valve **16** thereby reducing the absolute pressure of air supplied to first stage **11** of compressor **10**.

In the preferred embodiment of the invention, amplifier **46** may comprise a differential amplifier and summing circuit **45** may comprise a terminal or other network arrangement for summing current signals generated by sensors **34, 36** and **38** and reference signal generator **40**. Thus, the summation of the net current injected into the differential amplifier circuit from the current reference and the sensors has a magnitude and a direction which unbalances the differential amplifier in a direction to provide a suitable inlet valve opening signal or closing signal, depending on the direction of the unbalance.

In the preferred embodiment of this invention, the output from differential amplifier **46** is a voltage signal from one or the other of a pair of transistors depending on the direction of the unbalance indicated by the control signal. The voltage output from one of the transistors is employed to switch a Triac or other switching device to its conducting state in the valve opening circuit and the voltage output from the other transistor is employed to switch a Triac to its conducting state in the valve closing state. The Triacs in turn control the passage of current through the windings of a motor which rotates to open or close the inlet gas valve in a direction to reduce the unbalance in the differential amplifier by changing the sensed inlet air pressure.

In operation, compressor **10** is driven by a suitable prime mover (not shown) such as an electric motor. Compressed air is discharged from the compressor into receiver **19** until discharge pressure sensor **25** indicates that the set point pressure has been reached. When the set point pressure is reached, control **27** causes dump valve actuator **28** to move toward an open position thereby preventing passage of excess air into receiver **19** until the pressure in the receiver is reduced below the set point pressure of the main control.

Inlet gas temperature sensor **34** and interstage gas temperature sensor **35** sense the temperature of the gas admitted to the first and second stages respectively of compressor **10** and provide suitable temperature function current signals to summing circuit **45**. Likewise, inlet pressure sensor **35** provides a pressure function current signal to summing circuit **45** as does reference signal generator **40**. If the sum of the currents generated by the sensors is substantially equal to that generated by reference signal generator **40**, no control signal will be supplied to amplifier **46**. Preferably, a dead band is provided which will span a sufficient range of control signal magnitude to prevent hunting in the control characteristic. If, however, the deviation of the summed sensed function signals from the reference signal is greater than the predetermined dead band, a control signal will be passed to amplifier **46** having a direction and magnitude corresponding to the necessary change in inlet pressure that is required to again balance the operation of the system. From amplifier **46**, the valve opening or closing signal required for rebalancing the system will be passed to the appropriate valve opening or closing circuit to move the inlet valve actuator **49** in the proper direction to reduce the deviation from the desired operating condition.

For example, if for some reason, the interstage temperature should suddenly rise as might happen if the interstage cooling water temperature rose or its flow rate decreased, an unbalance between the reference signal and the sensed function signals would be reflected in a control signal from summing circuit **45** causing a valve opening signal to be passed to valve actuator **49** thereby preventing the compressor from surging by increasing the inlet pressure to first stage **11** and thus decrease the overall compressor pressure ratio. If the temperature of the inlet air to first stage **11** rises, as for example due to increase in ambient temperature, or if the inlet pressure should drop, the summing circuit will provide a control signal to amplifier **46** to unbalance it in a direction to open inlet gas valve **16** thereby reducing the pressure lift across compressor **10** and preventing a surge condition from occurring. Similarly, if the interstage temperature or inlet temperature drop, or the inlet pressure rises, differential amplifier **46** will provide a valve closing signal to actuator **49** to prevent overloading of the prime mover by reducing the mass flow through the compressor.

In accordance with this invention, there is provided a gas compression sytem having a control over inlet air admitted to the compressor which serves to compensate for both variations in inlet air temperature and pressure and also for variations in interstage conditions. The point of maximum prime mover load can be closely approached without damage of an accidental overload due to changing gas conditions at any point in the system. Likewise, the so-called "surge line" of the gas compression system may be closely approached without causing the compressor to go into a surge condition due to unexpected changes in ambient or interstage gas conditions. Therefore, the gas compression system of this invention can be operated at optimum conditions over a wider range of gas conditions than could be tolerated by prior systems. Consequently, a gas compression system embodying this invention can be operated at high capacity and with great operating efficiency.

While the invention has been described with reference to a preferred embodiment thereof wherein the gas compression system supplies compressed air, it will be understood that the principles of this invention are applicable to other systems embodying other gases and for other purposes, if desired.

I claim:

1. A gas compression system including a multistage gas compressor having a first gas compression stage and at least one following gas compression stage connected in series relation with said first stage, a prime mover connected to drive said compressor, an inlet passage for admitting inlet gas to the first stage of said gas compressor, a throttling valve for controlling flow of inlet gas to said first stage of said compressor, an outlet passage for discharging compressed discharge gas from said system, and a main control system for sensing discharge gas pressure and adjusting the conditions of operation of said system in accordance with the sensed discharge gas pressure to maintain a substantially predetermined discharge gas pressure, wherein the improvement comprises the combination of:

(a) inlet gas temperature sensing means disposed for sensing the temperature of inlet gas admitted to the first stage of said compressor, and for providing an inlet gas temperature signal functionally related thereto;

(b) interstage gas temperature sensing means for sensing the temperature of gas admitted to a following stage of said compressor, and for providing an interstage gas temperature signal functionally related thereto;

(c) inlet gas pressure sensing means disposed for sensing the pressure of inlet gas admitted to the first stage of said compressor, and for providing an inlet gas pressure signal functionally related thereto;

(d) means for summing said inlet and interstage temperature signals and said pressure signal to provide a summed function signal which is functionally related to each of the combined sensed conditions of operation of said gas compressor;

(e) reference signal means for providing a reference signal functionally related to a predetermined combination of each of said sensed conditions which results in a desired condition of operation of said gas compressor;

(f) means for detecting deviation of said summed function signal from said reference signal and for providing a control signal which is a function of said deviation; and

(g) means for adjusting said throttling valve in response to said control signal to reduce said deviation by varying the inlet gas pressure in a manner so as to shift the condition of operation of said gas compressor toward said desired condition, thereby tending to prevent overloading of said prime mover and surging of said compressor.

2. A gas compression system as defined in claim **1** wherein said compressor includes an interstage cooler between the first stage thereof and said following stage thereof; means to pass a cooling medium in through said interstage cooler in heat exchange relation with gas compressed from the first stage of said compressor to cool said compressed gas prior to entry into said following stage; and said interstage temperature sensor being located to sense the tempertaure of gas admitted to said following stage of said compressor after being cooled in said interstage cooler.

3. A gas compression system as defined in claim **1** wherein said inlet gas temperature signal generated by said inlet gas temperature sensing means is a nonlinear function corresponding to a predetermined nonlinear condition of operation of said gas compression system.

4. A method of operating a gas compression system having a multiple stage gas compressor driven by a prime mover to reduce overloading of said prime mover and surging of said compressor, which includes the steps of sensing the discharge gas pressure of said system and adjusting the condition of operation of said system in response to sensed discharge gas pressure to maintain a substantially constant predetermined discharge pressure, wherein the improvement comprises:

(a) sensing the temperature of gas admitted to a first stage of said compressor and generating an inlet gas temperature signal functionally related thereto;

(b) sensing the temperature of gas admitted to a stage of said compressor following said first stage thereof, and generating an interstage temperature signal functionally related thereto;

(c) sensing the pressure of gas admitted to the first stage of said compressor and generating a pressure signal functionally related thereto;

(d) generating a reference signal having a magnitude related to a desired condition of operation of said gas compression system;

(e) summing said inlet tempertaure signal, said interstage temperature signal, said inlet pressure signal and said reference signal to provide a control signal which is functionally related to the difference between the condition of operation of said gas compression system and the desired condition of operation thereof; and

(f) controlling the rate of flow of gas to the inlet of said first stage of said compressor in response to said control signal to change inlet gas pressure to said compressor in a manner to approach the desired condition of operation of said system, thereby reducing overloading of said prime mover and surging of said compressor.

5. A method of operating a gas compression system as defined in claim **4** including the step of cooling the compressed gas discharged from said first stage of said compressor prior to entry thereof into said following stage of said compressor, and sensing the temperature of the gas admitted to said following stage subsequent to cooling thereof so that the effect of cooling of said gas is sensed and compensated for in controlling the rate of flow of gas to the inlet of said first stage.

References Cited

UNITED STATES PATENTS

1,977,559	10/1934	Lewis et al.	266—30
2,632,307	3/1953	Massey et al.	230—114 X
2,696,345	12/1954	Hopper	230—115
2,929,547	3/1960	Koffel	230—115
2,938,536	5/1960	Ehrenberg	103—11 X
2,955,745	10/1960	Hunter	230—115
3,007,414	11/1961	Long et al.	230—115 X
3,027,904	4/1962	Silver	230—115 X
3,232,519	2/1966	Long	230—114 X

WILLIAM L. FREEH, *Primary Examiner.*

U.S. Cl. X.R.

230—23

PO-1050
(5/69)

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,441,200 Dated April 29, 1969

Inventor(s) Eugene L. Huesgen

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 2, line 46, for "operations" read -- operation --; line 48, for "multisage" read -- multistage --.
Column 3, line 28, for "scond" read -- second --.
Column 6, line 33, after "predetermined" insert -- desired --.

SIGNED AND
SEALED
MAR 10 1970

(SEAL)
Attest:

Edward M. Fletcher, Jr.
Attesting Officer

WILLIAM E. SCHUYLER, JR.
Commissioner of Patents