

SYSTEM OF ENERGY-EFFICIENT AND CONSTANT-PRESSURE PARALLEL-COUPLED FLUID-TRANSPORT MACHINES

5 BACKGROUND OF THE INVENTION

Field of the Invention

10 The present invention relates to a parallel-coupled fluid-transport
machinery, particularly to a system of energy-efficient and
constant-pressure parallel-coupled fluid-transport machines.

15 Description of the Related Art

For recent years, owing to the advance of technology, the industrial products are growing precise, delicate and miniature more and more. To mass-produce those precise, delicate and miniature products in low cost, the current high-tech factories grow in scale continuously, and the quality requirements thereof also become more and more strict, which is a challenge for the building companies of the high-tech factory and the facility engineers who maintain the high-tech factory, and which also drives the related personnel to zealously research solutions or develop new technologies to fulfill the requirements.

In the current high-tech factories, it is a basic requirement for

fluid-transport machinery to supply the gas and water of sufficient flow rate and stable pressure to every unit requiring them. When the factory is of smaller scale, merely a single fluid-transport machine is enough to supply the required flow rate. When only a single machine is used, the pressure stability can be easily and simply controlled according the operational manual provided by the manufacturer. However, in the current high-tech factories, the scale of the factory is enlarged continuously, and the fabrication procedures also become more and more flexible, in order to reduce the fabrication cost and to achieve the product diversification, and therefore, a single-machine fluid-transport system can no more meet the requirement. In order to supply the gas and water of sufficient flow rate and stable pressure to every unit requiring them and to shift the pressure and flow rate of gas and water as soon as possible and meet the new procedure's requirements when the fabrication procedures are changed, a system of multiple parallel-coupled fluid-transport machines is needed in the current high-tech factories. The operation of parallel-coupled fluid-transport machines is more complicated and crucial than that of a single fluid-transport machine, and just an imprudent operation may damage the machines. However, even the manufacturer cannot provide a standard operational procedure for the system of parallel-coupled transport-fluid transport machines. Thus, the factory-building companies and the facility-maintaining personnel can only depend on the experience accumulated in practical operation. The operational standard assumed in such a way is not only multitudinous and complicated but also neither analyzed theoretically nor verified experimentally. Therefore, it is hard to determine whether the assumed

operational standard can achieve the required security and rapidity. The system of multiple parallel-coupled fluid-transport machines consumes considerable energy; therefore, reducing the consumed energy thereof can benefit the fabrication cost very much. However, the objective of saving energy cannot be realized via the current operational standard, which is assumed from individual experience.

When the user purchases a centrifugal fluid-transport machine, such as a pump, a blower, or an exhaust fan, the manufacturer will provide the user with operational data, such as the performance curves shown in Fig.1. In Fig.1, the horizontal and vertical axes respectively represent flow rate and pressure. The performance curve of maximum usable rotation speed begins from the vertical axis and ends in some point of the diagram, which represents the critical usable point. The area located at the right side of the performance curve of maximum usable rotation speed is a region unsuitable to use. According the Affinity Law of fluid-transport machines, the relationships with respect to rotation speed (N), flow rate (Q), pressure (P), and power (HP) are shown below.

$$\left. \begin{aligned} \frac{Q_1}{Q_2} &= \frac{N_1}{N_2} \\ \frac{H_1}{H_2} &= \left(\frac{N_1}{N_2} \right)^2 \\ \frac{HP_1}{HP_2} &= \left(\frac{N_1}{N_2} \right)^3 \end{aligned} \right\} \text{-----}(1)$$

Based on the data of the performance curve of maximum usable rotation speed, and via theoretical analysis and calculation, the following

equations respectively relating pressure (P) with rotation speed (N) and relating pressure (P) with flow rate (Q) can be obtained.

$$P = C_1 N^2 + C_2 N + C_3 \text{ -----}(2)$$

$$P = C_1 Q^2 + C_2 Q + C_3 \text{ -----}(3)$$

5 Via equations (2) and (3), the performance curves of the rotation speeds lower than the maximum usable speed can be obtained, as the dashed curves shown in Fig.1. Further, the equal-efficiency curves and the equal-power curves are also plotted in Fig.1.

There are not many patents about the system of multiple
10 parallel-coupled fluid-transport machines proposed before. The Taiwan Patent No.506683 “Parallel-coupled Electrical Fans” discloses: a system of multiple parallel-coupled electrical fans, wherein multiple electrical fans are installed in parallel to a one-piece frame; however, it does not mention how to control the parallel-coupled electrical fans at all. As to the
15 method of controlling the fluid-transport machine, the Taiwan Patent No.I225908 “Method of Controlling a Pump System” discloses: a method of utilizing automatic control and operational parameters of a centrifugal pump to pump fluid to an outlet; however, this patent is confined to the control of a single centrifugal pump, wherein pump rotation speed, water
20 temperature, pressures at the inlet and outlet of the pump, and the pressure difference thereof detected by the sensors are compared with the pre-stored data, and then, the pump is adjusted according to the comparing result. The Taiwan Patent No.M253699 “Devices of Controlling a Pump System” discloses: a variety of devices of controlling

a single centrifugal pump, which is similar to the previous patent. As shown in their claims, the abovementioned pre-stored data is not deduced from the theories but acquired via arranging the data gathered from practical operations; such a method advantages in that the states of a pump, which are to be the basis of control, can be easily and rapidly obtained; however, there are two constrains on such a method: one is that it cannot apply to the system of multiple parallel-coupled machines but can only apply to a single machine; the other is that the control range is limited by the pre-stored data. About energy efficiency, whether energy can be saved thereby is an unknown.

Accordingly, via theoretical analysis and experimental verification, the present invention proposes a system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines and the operational method thereof, which not only can achieve the secure, rapid, and energy-efficient operation of the system of multiple parallel-coupled fluid-transport machines but also can unify and simplify the operational standard thereof, and the building cost thereof can also be lowered.

Summary of the invention

Sequentially via the following theoretical analysis, equation deduction, and embodiment discussion, the system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines of the present invention will be described in detail below. The flow rate in the load piping system is very important in the study of the system of multiple

parallel-coupled fluid-transport machines; therefore, the distribution of the flow rate in the load piping system is to be calculated firstly.

The piping system shown in Fig.2 is to be used as an exemplification of the load piping of the system of multiple parallel-coupled fluid-transport machines. As shown in Fig.2, the piping system has 9 pipe sections (from pipe section 1 to pipe section 9) and 10 nodes (from node 1 to node 10, including: fluid inlets, fluid outlets and confluence points, i.e. the ends of all the pipe sections). Nodes 1, 2, 5, 6 are the inlets of branch pipes, and nodes 9, 10 are the total outlets. Pipe sections 8, 9 respectively have fluid-transport machines, and the arrows thereof are not necessarily the real flow direction but the assumptive direction used in the calculation by programs. In the system, the value to be calculated is the flow rate Q_n in each pipe section, wherein the subscript n denotes the serial number of the pipe section. As there are 9 pipe sections, there are nine unknown values $Q_1, Q_2, \dots, Q_9$, which need nine independent linear equations to solve. According to the conservation of the flow rate at the nodes, the continuity equations can be established; for example, at node 3, the flow rate in pipe section 3 is equal to the sum of the flow rates in pipe sections 1 and 2, and it can be mathematically expressed as equation (4):

$$Q_1 + Q_2 - Q_3 = 0 \quad (4)$$

Similarly, at nodes 4, 7, 8, there are also three flow-rate-conservation equations (5), (6), (7):

$$Q_3 + Q_4 - Q_5 = 0 \quad (5)$$

$$Q_5 + Q_6 - Q_7 = 0 \quad (6)$$

$$Q_7 - Q_8 - Q_9 = 0 \quad (7)$$

As the conservation of flow rate does not exist at nodes 1, 2, 5, 6, 9, and 10, there is no flow-rate-conservation equation to implement solving flow rates.

5 The other five equations for solving flow rates can be acquired from the conservation of energy. For example, pipe section 3 has node 3 and node 4 respectively at two ends thereof, and when the fluid flows from node 3 to node 4, it means that the total pressure at node 3 is greater than that at node 4; the total pressures at nodes 3 and 4 are respectively
10 denoted by P_3 and P_4 ; owing to friction and the piping structure, there is a loss of total pressure when the fluid flows from node 3 to node 4; the total pressure loss is denoted by ΔP_3 , and then,

$$\Delta P_3 = P_3 - P_4 \quad (8)$$

The total pressure loss in pipe section 3 can also be mathematically
15 expressed as $K_3 Q_3^2$ (it will be explained later), wherein K_3 is the coefficient of the total pressure loss in pipe section 3; then, equation (8) can be written as:

$$K_3 Q_3^2 = P_3 - P_4 \quad (9)$$

As each pipe section has its own structure and devices, each pipe section
20 has different total pressure loss coefficient K_n , wherein the subscript is used to denote the serial number of the pipe section. The left portion of equation (9) has an unknown Q_3 , and the right portion has two unknowns P_3 and P_4 ; thus, increasing one equation is at the cost of adding two

redundant unknowns. The total pressures of all the inlets and outlets (i.e. nodes 1, 2, 5, 6, 9 and 10) can be supposed to be known. When the fluid enters into the load piping from node 1 and flows through pipe sections 1, 3, 5, 7 and 9 and then exits from node 10, according to the principle of energy conservation, which has been used in equation (8),

$$\Delta P = P_1 - P_{10} \quad (10)$$

wherein ΔP is the sum of the total pressure loss in every pipe section and the total pressure change caused by the fluid-transport machine. Thus,

$$\Delta P = \Delta P_1 + \Delta P_3 + \Delta P_5 + \Delta P_7 + \Delta P_9 - \Delta P_{s9} \quad (11)$$

wherein ΔP_{s9} is the total pressure change induced by the fluid-transport machine when the fluid flows through pipe section 9, and how to deal with ΔP_{s9} will be discussed later. Substitute the expression of ΔP in equation (11) for ΔP in equation (10), and rearrange the terms as follows:

$$\Delta P_{s9} - \Delta P_9 - \Delta P_7 - \Delta P_5 - \Delta P_3 - \Delta P_1 = P_{10} - P_1 = C_1 \quad (12)$$

P_{10} and P_1 are known and usually the pressures of the exterior environments; therefore, C_1 is a constant. Transform ΔP into the form of KQ^2 to obtain:

$$\Delta P_{s9} - K_9 Q_9^2 - K_7 Q_7^2 - K_5 Q_5^2 - K_3 Q_3^2 - K_1 Q_1^2 = C_1 \quad (13)$$

Similarly, the following equations can also be obtained:

$$\Delta P_{s9} - K_9 Q_9^2 - K_7 Q_7^2 - K_5 Q_5^2 - K_3 Q_3^2 - K_2 Q_2^2 = P_{10} - P_2 = C_2 \quad (14)$$

$$\Delta P_{s9} - K_9 Q_9^2 - K_7 Q_7^2 - K_5 Q_5^2 - K_4 Q_4^2 = P_{10} - P_5 = C_3 \quad (15)$$

$$\Delta P_{s9} - K_9 Q_9^2 - K_7 Q_7^2 - K_6 Q_6^2 = P_{10} - P_6 = C_4 \quad (16)$$

$$\Delta P_{s8} - K_8 Q_8^2 - K_7 Q_7^2 - K_6 Q_6^2 = P_9 - P_6 = C_5 \quad (17)$$

So far, from equations (4)~(7) and from equations (13) to (17), there are totally nine linear independent equations. In addition to nine unknowns of flow rates, there are still two unknowns ΔP_{s8} and ΔP_{s9} existing among those equations, which will be solved from the performance curves of the fluid-transport machines.

The abovementioned quadratic polynomial equation (3) can match the reality very well; therefore, the performance curve of the fluid-transport machine at a given rotation speed can be mathematically expressed as:

$$\Delta P_{s8} = C_{s1} Q_8^2 + C_{s2} Q_8 + C_{s3} \quad (18)$$

$$\Delta P_{s9} = C_{s1}' Q_9^2 + C_{s2}' Q_9 + C_{s3}' \quad (19)$$

wherein C_{s1} , C_{s2} , C_{s3} , C_{s1}' , C_{s2}' , and C_{s3}' are all constants, and the subscript S denotes the fluid-transport machine. Except ΔP_{s8} and ΔP_{s9} , each term in left sides of the equal signs of equations (13)~(17) is the total pressure loss coefficient multiplied by the flow rate squared. In order to add ΔP_{s8} and ΔP_{s9} into the calculation of the equations, via formula method, equations (18) and (19) are rewritten as:

$$\Delta P_{s8} = C_{s1} G_8^2 + C_{s4} \quad (20)$$

$$\Delta P_{s9} = C_{s1}' G_9^2 + C_{s4}' \quad (21)$$

wherein $G_8 = Q_8 + \frac{C_{s2}}{2C_{s1}}$, $G_9 = Q_9 + \frac{C_{s2}'}{2C_{s1}'}$, $C_{s4} = C_{s3} - \frac{C_{s2}^2}{4C_{s1}}$, and

$C_{s4}' = C_{s3}' - \frac{C_{s2}^2}{4C_{s1}}$. Substitute the expression of ΔP_{s9} in equation (21) for

ΔP_{s9} in equation (13), and rearrange the terms as follows:

$$C_{s1}' G_9^2 - K_9 Q_9^2 - K_7 Q_7^2 - K_5 Q_5^2 - K_3 Q_3^2 - K_1 Q_1^2 = C_1 - C_{s4}' \quad (22)$$

Similarly, the expression of ΔP_{s8} and ΔP_{s9} in equations (20) and

(21) can also substitute for ΔP_{s8} and ΔP_{s9} in equations (14)~(17). It can be found: in addition to the unknowns of flow rates, there are still two unknowns G_8 and G_9 existing in the left sides of the equal signs, and therefore, further two equations are needed, i.e. increasing one fluid-transport machine will increase one unknown also. From the process of the formula method, the relationships of G_8 and G_9 versus the flow rates of the corresponding pipe sections can be derived and mathematically expressed as:

$$G_8 - Q_8 = \frac{C_{s2}}{2C_{s1}} \quad (23)$$

$$G_9 - Q_9 = \frac{C_{s2}'}{2C_{s1}} \quad (24)$$

So far, the system of equations has 11 equations and 11 unknowns, and thus, the flow rates of all the pipe sections can be solved. The K_n ($n=1, 2, \dots, 9$) in equations (13)~(17) are all known values, and they are the coefficients of the total pressure loss, which includes: the loss caused by the friction between the fluid and the pipe, and the minor loss caused by the devices and parts installed to the pipe; therefore, the K values can be obtained via referring to the pipe loss coefficients.

When the fluid-transport machines of the same specification are

coupled in parallel, the flow rate in the performance curve will also increase as many times as the fluid-transport machine does. Fig.3 shows the performance curves of the systems having different numbers of parallel-coupled fluid-transport machines operating at a fixed speed. As shown in Fig.3, at an identical pressure, the flow rate provided by two parallel-coupled fluid-transport machines is the double of that provided by a single fluid-transport machine. Similarly, at an identical pressure, the flow rate provided by three parallel-coupled fluid-transport machines is the triple of that provided by a single fluid-transport machine. In other words, the performance curves change with the quantities of parallel-coupled fluid-transport machines. If the performance curve of a single fluid-transport machine is mathematically expressed as:

$$P = C_{s1}Q^2 + C_{s2}Q + C_{s3} \quad (25)$$

wherein C_{s1} , C_{s2} , and C_{s3} are constants. Let Q_n ($n=1, 2, 3, \dots, n$) denote the integrated flow rate of the system of the parallel-coupled fluid-transport machines with n denoting the number of the parallel-coupled fluid-transport machines. When n fluid-transport machines are coupled in parallel, the fluid rate thereof will increase by n times, i.e. $Q_n = n \cdot Q$; substitute it into equation (25) to obtain the

following equation:

$$P = \frac{C_{s1}}{n^2} Q_n^2 + \frac{C_{s2}}{n} Q_n + C_{s3} \quad (26)$$

which mathematically expresses the performance curve of the single equivalent machine of the system of multiple parallel-coupled fluid-transport machines.

For a complicated piping system, if the setting of the system is

unchanged, the total pressure loss of the system will be proportional to the total flow rate squared. A system impedance curve can be formed via plotting the total pressure loss P against the total flow rate Q and can be mathematically expressed as:

$$\Delta P_{sys} = K_{sys} Q_{sys}^2 \quad (27)$$

wherein K_{sys} is a constant, and Q_{sys} is the total flow rate of the system, and ΔP_{sys} is the total pressure loss of the system. Therefore, with the system conditions unchanged, if the pressures at some flow rates are known, the system impedance curve can be acquired thereby.

Fig.4 shows a system impedance curve and a performance curve of a single fluid-transport machine and an intercept point of them. The pressure and the flow rate corresponding to the intercept point are exactly the total pressure loss and the total flow rate of the system respectively, and therefore, the intercept point is called the operational point. In the piping system of a single fluid-transport machine, the total pressure loss of the system is equal to the pressure provided by the fluid-transport machine, and the total flow rate of the system is equal to the flow rate of the fluid-transport machine. If the mathematic expressions of the performance curve of the fluid-transport machine and the system impedance curve are known, substitute equation (27) into the equation of the performance curve to obtain:

$$C_{S1} Q_{sys}^2 + C_{S2} Q_{sys} + C_{S3} = K_{sys} Q_{sys}^2 \quad (28)$$

and then, obtain the solutions of Q_{sys} —the flow rate at the intercept point (the operational point) of the those two curves—via the formula method:

$$Q_{sys} = \frac{-C_{s2} \pm \sqrt{C_{s2}^2 - 4(C_{s1} - K_{sys})C_{s3}}}{2(C_{s1} - K_{sys})} \quad (29)$$

For a fluid-transport machine installed in a piping system and operating at a fixed rotation speed, the flow rate thereof can be solved from equation (29). If the solved flow rate is not the expected value, the flow rate can be adjusted via changing the rotation speed of the fluid-transport machine. When the rotation speed of the fluid-transport machine is changed, the performance curve thereof also changes, and the operational point too. The performance curve of the fluid-transport machine can be worked out via the calculation according to the Affinity Law, and the relationships of flow rate versus rotation speed, pressure versus rotation speed, and power versus rotation speed have been shown in equation (1).

Fig.5 shows the performance curves of a fluid-transport machine when it operates at the rotation speeds of 1750 rpm and 2275rpm, wherein Point A and Point B are the operational points for different rotation speeds in the fixed system. From equation (27), the following relationship can be obtained:

$$\frac{P_A}{P_B} = \frac{K_{sys} Q_A^2}{K_{sys} Q_B^2} = \frac{Q_A^2}{Q_B^2} \quad (30)$$

According to equation (1), the relationship between Point A in the performance curve of 1750rpm and Point C in the performance curve of 2275rpm can be written as:

$$\frac{Q_A}{Q_C} = \frac{N_{1750}}{N_{2275}} \quad (31)$$

$$\frac{P_A}{P_C} = \frac{N_{1750}^2}{N_{2275}^2} \quad (32)$$

Substituting equation (31) into equation (32) can obtain:

$$\frac{P_A}{P_C} = \frac{Q_A^2}{Q_C^2} \quad (33)$$

Via comparing equation (30) and equation (33), it is found: that Point B is
 5 exactly Point C. Therefore, when Operational Point A is known, Operational Point B, i.e. the intercept point of another performance curve and the system impedance curve, can be worked out with the Affinity Law.

When the pressure and the flow rate of Point B is the target
 10 pressure and the target flow rate of the system yet with the rotation speed of the fluid-transport machine being unknown, the rotation speed can be worked out as follows:

Firstly, according to equation (27), the parameter K_{sys} is obtained by:

$$K_{sys} = \frac{P_B}{Q_B^2} \quad (34)$$

15 Suppose that the performance curve of the fluid-transport machine operating at the rotation speed of 1750rpm can be mathematically expressed as equation (25). Substitute the coefficients of equation (25) and equation (34) into equation (29) to obtain the flow rate of the system Q_A . Then, according to equation (1), the rotation speed can be obtained
 20 by:

$$N_{sys} = N_{1750} \frac{Q_B}{Q_A} \quad (35)$$

Thus, the target flow rate can be obtained via adjusting the rotation speed of the fluid-transport machine to N_{sys} . According to equation (1), the

performance curve of the adjusted rotation speed can be mathematically expressed as:

$$\frac{N_{1750}^2}{N_{sys}^2} P = C_{s1} \left(\frac{N_{1750}}{N_{sys}} Q \right)^2 + C_{s2} \left(\frac{N_{1750}}{N_{sys}} Q \right) + C_{s3} \quad (36)$$

Rearrange it to obtain:

$$5 \quad P = C_{s1} Q^2 + C_{s2} \frac{N_{sys}}{N_{1750}} Q + \frac{N_{sys}^2}{N_{1750}^2} C_{s3} \quad (37)$$

According to the principles mentioned above, and based on the data shown in Fig.6 and Fig.7, the performance curve of Fig.8 can be mathematically expressed as:

$$P = -0.8889Q^2 + 6.6667Q + 700 \quad (38)$$

- 10 In this exemplification, three fluid-transport machines of the same specification are coupled in parallel. According to equation (26), the performance curve of the single equivalent fluid-transport machine can be mathematically expressed as:

$$P = -0.0988Q^2 + 2.2222Q + 700 \quad (39)$$

- 15 Via equation (27), K_{sys} is obtained by:

$$K_{sys} = \frac{563}{40^2} = 0.352 \quad (40)$$

Substitute the coefficients of the equation (39) and the constant of equation (40) into equation (29) to obtain:

$$Q_{sys} = \frac{-2.2222 - \sqrt{2.2222^2 - 4(-0.0988 - 0.352)700}}{2(-0.0988 - 0.352)} = 41.9 \text{ (cfs)}. \text{ As shown in}$$

- 20 Fig.9, the worked-out total flow rate of the system is 41.9cfs; however, the total flow rate required by the system is only 40cfs; therefore, the rotation speed of the fluid-transport machine should be lowered. Suppose

equation (38) is of the performance curve of the rotation speed of 1750rpm. The desired rotation speed can be obtained according to equation (35) by:

$$N_{sys} = N_{1750} \frac{Q_B}{Q_A} = 1750 \frac{40}{41.9} \approx 1670(\text{rpm}).$$

When the rotation speed is reduced to 1670rpm, the performance curve can be obtained according to equation (37) and mathematically expressed as:

$$P = -0.8889Q^2 + 6.3619Q + 637.46 \quad (41)$$

Adopt the performance curve mathematically expressed by equation (41); the recalculated total flow rate will be 40 *cfs*, and the flow rates of all outlets are almost identical, and the pressures thereof are also uniform, as shown in the portion for three parallel-coupled machines of Fig.10. According to the same method, the rotation speed required by two parallel-coupled machines can also be worked out, and the rotation speed thereof should be raised to 1850rpm. As shown in the portion for two parallel-coupled machines of Fig.10, the flow rates of all outlets are also almost identical, and the errors of the pressures are within only 2%.

As to the equal-efficiency curves in Fig.1, they cannot be deduced theoretically but are obtained experimentally from practical operation by the manufacturer, and those data is no more further analyzed but will be used directly.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig.1 is a diagram showing the data of a centrifugal fluid-transport

machine.

Fig.2 is a diagram showing the system of multiple parallel-coupled fluid-transport machines and the load piping system.

Fig.3 is a diagram showing the performance curves of a single fluid-transport machine, the system of two parallel-coupled fluid-transport machines, and the system of three parallel-coupled fluid-transport machines.

Fig.4 is a diagram showing the performance curve of a fluid-transport machine and the system impedance curve.

Fig.5 is a diagram showing the performance curves of a fluid-transport machine operating at the rotation speeds of 1750rpm and 2275rpm.

Fig.6 is a diagram showing the structure of a load piping system.

Fig.7 is a diagram showing the data of a working fluid and the data of all pipe sections.

Fig.8 is a diagram showing the performance curves of a fluid-transport machine operating at the rotation speed of 1750rpm.

Fig.9 is a diagram showing the flow rate distribution in each of the outlets when the fluid-transport machines operate at the rotation speed of 1750rpm.

Fig.10 is a diagram showing the flow rate distribution in each of the outlets when the fluid-transport machines operate at the rotation speed of 1670rpm.

Fig.11 is a diagram showing the flow rate and the pressure required by the system and the performance curves of the system of two parallel-coupled fluid-transport machines and the system of three parallel-coupled fluid-transport machines.

Fig.12 is a diagram showing the performance curves of the system of two parallel-coupled fluid-transport machines and the system of three parallel-coupled fluid-transport machines.

Fig.13 is a diagram showing the data relating to pressure and flow rate and containing equal-efficiency curves

Fig.14 is a diagram showing the convention operational method and the operational method of the present invention.

Fig.15 is a diagram showing the layout of the constituent parts of the present invention.

Fig.16 is a diagram showing the architecture of the system control.

DETAILED DESCRIPTION OF THE INVENTION

In practical operation, the time allowed to determine operational parameters is very short; therefore, it is hard to utilize the equations deduced above to calculate the operational parameters and then apply the worked-out parameters to control the system; thus, the required operational parameters are worked out beforehand, and then, the work-out parameters are directly applied to the system. The embodiments described below are to exemplify that the results worked out via the abovementioned equations are directly applied to the practical cases.

The embodiment to be discussed below supposes that the flow rate required by the system is Q_T , and the pressure of the system is a constant pressure P_T , and the maximum flow rate a single pump can

provides is Q_1 , and the maximum pressure a single pump can provide is P_1 . When the system uses multiple pumps, the embodiment also supposes that those pumps are of the same specification of the same manufacturer in principle. From the abovementioned basic data, it is known: if $Q_T > Q_1$,
5 a single pump cannot meet the requirement of flow rate, and if $P_1 > P_T$, it can meet the requirement of pressure.

In the conventional operational method, the system impedance can be calculated from the piping layout and the flow rate of each pipe section, and the system impedance curve can be formed via plotting the system
10 impedance against the flow rate, and the conventional operational method comprises the following procedures:

- (1) Turn on a first pump; however, as shown in Fig.11, the performance curve of the single pump (the solid curve) cannot intercept the system impedance curve at Operational Point O of $P = P_T$ and $Q = Q_T$;
- 15 (2) Turn on a second pump with the operational conditions the same as that of the first pump; the performance curve of two parallel-coupled pumps (---) intercepts the system impedance curve at Point X, which exceeds the system requirement; then, the rotation speeds of those two pumps are lowered, and thus, the performance curve of two
20 parallel-coupled pumps (- • -) can intercept the system impedance curve at Operational Point O;
- (3) If a third pump is also turned on, as shown in Fig.12, via appropriately adjusting the rotation speed of each pump similar to that used in the case of two parallel-coupled pumps, the performance curve of three

parallel-coupled pumps can also intercept the system impedance curve at Operational Point O;

- (4) Keep on increasing the number of operating pumps, via appropriately adjusting the rotation speed of each pump similarly, the performance curve of multiple parallel-coupled pumps can also intercept the system impedance curve at Operational Point O;
- (5) Thus, the system requirement can be satisfied with only two parallel-coupled pumps or more.

However, the conventional operational method leaves the following problems to be solved: how many pumps should be parallel coupled so that the system can be the most energy-efficient, and how to adjust the number of parallel-coupled pumps to achieve the optimal state when the system impedance curve is changed by the change of the required flow rate, which are also the subjects the present invention desires to overcome.

From the pump theory, it is known: when the output flow rate of each pump is the same, energy saving is more likely to achieve. For example, the case of $Q_1 = Q_2 = Q_3$ is more energy-efficient than the case of $Q_1 \neq Q_2 \neq Q_3$. Therefore, making the flow rate output by each pump be equal will be the first task intended to do.

To enable the flow rate of each pump to be equal, power meters are respectively installed to all the pumps, and according to the values detected by the power meters, the flow rate of each pump can be obtained by:

$$HP = \frac{P \times Q}{\text{constant}} \quad (42)$$

wherein P is the system pressure and maintained unchanged; the constant is the product of the pump efficiency and the unit conversion factor. The efficiency should be identical when the states of all the pumps are the same. Once the power is known, the flow rate will be known also. In other words, if the power of each pump is identical, i.e. $HP_1 = HP_2 = HP_3$, $Q_1 = Q_2 = Q_3$. Flow meters can also be installed to the pumps, and the flow rate can be directly detected; however, the error of the flow meter is great, and the response thereof is also slow; therefore, using power meters is a more suitable measure.

The abovementioned method can achieve a better energy-efficient effect when multiple pumps are coupled in parallel, but it cannot determine how many pumps should be parallel coupled to achieve the best energy-efficient effect yet; for example, it cannot determine which one of $Q_1 = Q_2 = Q_3$ and $Q_1 = Q_2$ is better.

Fig.12 shows the performance curve of two parallel-coupled pumps and the performance curve of one of those two parallel-coupled pumps, wherein via flow meters, those two pumps are adjusted so that $Q_1 = Q_2 = \frac{Q_T}{2}$ and $P_1 = P_2 = P_T$. The dashed curve is the performance curve of two parallel-coupled pumps, and appropriately adjusting the rotation speed of each pump can enable the performance curve of two parallel-coupled pumps to intercept the system impedance curve at Operational Point O. The solid curve is the performance curve of one of

those two parallel-coupled pumps, and Point O_2 thereof has the flow rate of $\frac{Q_T}{2}$ and the pressure of P_T . Fig.12 also shows the performance curve of three parallel-coupled pumps and the performance curve of one of those three parallel-coupled pumps, wherein via flow meters, those three pumps are adjusted so that $Q_1 = Q_2 = Q_3 = \frac{1}{3}Q_T$ and $P_1 = P_2 = P_3 = P_T$. The dot-dashed curve is the performance curve of three parallel-coupled pumps, and similarly, appropriately adjusting the rotation speed of each pump can enable the performance curve of three parallel-coupled pumps to intercept the system impedance curve at Operational Point O. The performance curve of one of those three parallel-coupled pumps is denoted by the other solid curve, and Point O_3 thereof has the flow rate of $\frac{1}{3}Q_T$ and the pressure of P_T . Both the measure of two parallel-coupled pumps and the measure of three parallel-coupled pumps can achieve the pressure P_T and the flow rate Q_T required by the system. The pressures provided by the pumps of both the measures are all equal to P_T ; however, the flow rates provided by the pumps of both the measures are respectively $\frac{Q_T}{2}$ and $\frac{Q_T}{3}$, and Point O_2 is different from Point O_3 .

As mentioned above, the manufacturer provides the equal-efficiency curves for each specification of pump. Plotting the equal-efficiency curves on Fig.12 will obtain Fig.13. The hyperbolic dashed curves on Fig.13 are respectively the equal-efficiency curves of 70% efficiency (η), 60% efficiency (η) and 50% efficiency (η), which are plotted directly from the data provided by the manufacturer, and the

bottom points thereof are interconnected to form a dot-dashed curve. As there is the constraint of constant pressure P_T , Point O_2 , Point O_3 , and Point O are all on the same horizontal line. In Fig.13, Point O_2 is on the equal-efficiency curve of 60% efficiency (η), and Point O_3 is on the equal-efficiency curve of about 65% efficiency (η). The total output power of the parallel-coupled pumps $(HP)_T$ is the product of $(HP)_{\text{SINGLE PUMP}}$ and the number n of parallel-coupled pumps:

$$(HP)_T = HP_{\text{SINGLE PUMP}} \times \text{the number of pumps } (n) \quad (43)$$

wherein

$$(HP)_{\text{SINGLE PUMP}} = \frac{P_T \times Q}{\text{constant} \times \eta(\%)} \quad (44)$$

Thus,

$$(HP)_{T_2} = \frac{P_T \times \frac{1}{2} Q_T}{\text{constant} \times (60\%)} \times 2 = \frac{P_T \times Q_T}{\text{constant} \times (60\%)} \quad (45)$$

$$(HP)_{T_3} = \frac{P_T \times \frac{1}{3} Q_T}{\text{constant} \times (65\%)} \times 3 = \frac{P_T \times Q_T}{\text{constant} \times (65\%)} \quad (46)$$

From equations (45) and (46), it is known: $(HP)_{T_3}$ is smaller than $(HP)_{T_2}$, i.e. the total output power of three parallel-coupled pumps is smaller than that of two parallel-coupled pumps; in other words, the system of three parallel-coupled pumps is more energy-efficient than the system of two parallel-coupled pumps.

When the system flow rate is changed to meet the requirement of fabrication, the system impedance curve will also change. When the flow

rate increases, the system impedance curve will shift right, and when the flow rate decreases, the system impedance curve will shift left. When the flow rate increases, there are two measures to deal with it: (1) do not increase the number of pumps but raise the rotation speed of the existing
5 pumps, (2) do not raise the rotation speed of the pumps but increase the number of pumps (In this case, the rotation speed may even need to decrease). According to the abovementioned principles, the total powers of those two measures can be respectively worked out to determine which measure has smaller power consumption and then determine which
10 measure is to be adopted. When the flow rate decreases, there are also two measures to deal with it: (1) do not decrease the number of pumps but lower the rotation speed of the existing pumps, (2) do not lower the rotation speed of the pumps but decrease the number of pumps (In this case, the rotation speed may even need to increase). Similarly, the total
15 powers of those two measures can be respectively worked out to determine which measure has smaller power consumption, and which measure is to be chosen depends on which measure consumes less power.

When the system impedance curve is changed, decreasing, increasing, or maintaining the number of pumps needs an operation
20 action to change the rotation speeds of the pumps, which is very critical to the high-tech factory. It is well known: high-tech factories take maintaining the stability of operational environment very seriously; inappropriate change of operational environment will endanger the fabricating products. It is often reported by the news: a short instability of
25 power supply instantly harms local manufacturers and causes considerable loss. According to the principle of the performance curve,

when the rotation speed of pumps is changed, the pressure of fluid supply would be changed also. Similar to the change of power supply voltage, the change of pressure also has a profound influence. Just a slight imprudence may cause a serious damage. For the current system, the time
5 allowed to adjust the system is very short; further, after the pressure is changed, the system requires response time to resume the stability of pressure. In the conventional operational method, when the system needs to increase flow rate, the rotation speed is often firstly raised so steeply that the pressure is found exceeding the set pressure very soon; then, the
10 rotation speed is reduced so steeply that the pressure is found lower than the set pressure very soon again; then, the rotation speed is raised once more. After a period of repeating the abovementioned process, the system finally assumes a stable pressure performance at the set pressure. The relationship of pressure (P) versus time (T) of the abovementioned
15 process is shown by the dashed curve of Fig.14.

Pressure is the driving force of fluid. The inappropriate change of pressure will cause the sudden change of the flow rate supply at the outlets, which will certainly bring about damage on the products and always besets the manufacturers, the operators and the maintenance
20 personnel. In the conventional technologies, when the requirement of flow rate is decreased, how to lower the rotation speed of pumps or how many pumps is to be turned off is an unknown; therefore, the measure to deal with the change is often not to change, i.e. the operational conditions is maintained unchanged, but the surplus fluid is recycled. It is obviously
25 spending precious energy in vain.

Accordingly, the present invention proposes a scheme to rapidly

and energy-efficiently deal with the abovementioned flow rate change. From those discussed above, it is known: when the system impedance curve is changed (i.e., the required flow rate is changed), the system and method proposed by the present invention can quickly work out the optimal number of operating pumps and the optimal rotation speed thereof. Thus, the rotation speed of pumps will be adjusted to be slightly less than the optimal rotation speed. When the pressure change is confirmed, the pumps will be slightly adjusted once more to operate at the optimal rotation speed. The relationship of pressure (P) versus time (T) of the abovementioned process is shown by the solid curve of Fig.14. Thereby, the fluid pressure of the system will not rise and fall repeatedly, and the flow supply at the outlets will no more increase and decrease alternately. Therefore, the present invention advantages the quality stability very much.

According to the embodiments of the present invention, the system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines of the present invention comprises: multiple variable-frequency centrifugal fluid-transport machines (61), multiple power meters (62), multiple pressure gauges (63), multiple flow meters (64), multiple controllers (65) and a load piping system (66), as shown in Fig.15. The functions of those constituent parts, the interactions thereof and the control procedures are shown in Fig.16 a diagram showing the control architecture of the system of the present invention.

In conclusion, the present invention proposes a system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines, wherein the data required by the parallel-coupled

fluid-transport machines is worked out via theoretical analysis and equation deduction; the worked-out data is used to control the system of parallel-coupled fluid-transport machines so that the system operation can be secure and quick. Further, the present invention can work out the
5 output powers of the system of different numbers of parallel-coupled fluid-transport machines and then can determine the number of operating fluid-transport machines, which can achieve the best energy-efficient performance according to the output powers thereof. The present invention is distinct from the abovementioned published patents;
10 therefore, the novelty and non-obviousness of the present invention is doubtless.

What is claimed is:

1. A system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines, comprising: performance curves of said system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines, system impedance curves of loads,
5 equal-efficiency curves, and multiple branch piping systems coupled in parallel, wherein each said parallel-coupled branch piping system further comprises: a variable-frequency centrifugal fluid-transport machine, power meters, pressure gauges, flow meters, and a controller.
- 10 2. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 1, wherein said performance curves of said system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines are worked out according to the relationships of the flow rate and the
15 pressure of said parallel-coupled fluid-transport machine.
3. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 1, wherein said system impedance curves of loads are worked out with the relationships of the flow rate and the impedance deduced from the load piping layout.
- 20 4. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 1, wherein said equal-efficiency curves are derived from the data provided by the manufacturer and stored in the form of a database containing the function relationships between flow rate and pressure.
- 25 5. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 1, wherein said

variable-frequency centrifugal fluid-transport machines are formed of variable-frequency centrifugal pumps.

6. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 1, wherein said
5 variable-frequency centrifugal fluid-transport machines are formed of variable-frequency centrifugal blowers.

7. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 1, wherein said
10 variable-frequency centrifugal fluid-transport machines are formed of variable-frequency centrifugal exhaust fans.

8. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 1, wherein said pressure
gauges are used to detect the pressure of fluid-transport pipes in order to maintain the state of constant pressure.

15 9. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 1, wherein said power
meters are used to measure the power output by said variable-frequency centrifugal fluid-transport machines, and the flow
rate can be obtained via substituting the measured power into the
20 equation: $\text{flow rate} = (\text{power} \times \text{constant}) / (\text{pressure})$, and said constant is a unit conversion factor.

10. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 1, wherein said flow
meters are used to measure the flow rate output by said
25 variable-frequency centrifugal fluid-transport machines in order to

guarantee that each said machine output an identical flow rate.

11. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 2, wherein said controller is formed of a programmable logic controller and has built-in data of said performance curves of said system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines in order to execute related calculation.

12. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 3, wherein said controller is formed of a programmable logic controller and has built-in data of said system impedance curves of loads in order to execute related calculation.

13. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 4, wherein said controller is formed of a programmable logic controller and has built-in data of said equal-efficiency curves in order to execute related calculation.

14. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 2, wherein said controller is formed of a computer and has built-in data of said performance curves of said system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines in order to execute related calculation.

15. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 3, wherein said controller is formed of a computer and has built-in data of said system impedance curves of loads in order to execute related calculation.

16. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 4, wherein said controller is formed of a computer and has built-in data of said equal-efficiency curves in order to execute related calculation.
- 5 17. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 11, wherein according to calculation results, said programmable logic controller controls the rotation speed of a variable-frequency centrifugal fluid-transport pump, or a variable-frequency centrifugal fluid-transport blower, or a
10 variable-frequency centrifugal fluid-transport exhaust fan.
18. The system of energy-efficient and constant-pressure parallel-coupled fluid-transport machines according to claim 14, wherein according to calculation results, said computer controls the rotation speed of a variable-frequency centrifugal fluid-transport pump, or a
15 variable-frequency centrifugal fluid-transport blower, or a variable-frequency centrifugal fluid-transport exhaust fan.